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JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. 49. AUGUST 1937. No. 3

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THE FUNDAMENTALS OF PARALLEL OPERATION OF DIRECT AND ALTERNATING CURRENT GENERATORS.

BY LIEUTENANT COMMANDER LOUIS DRELLER, U. S. NAVY, MEMBER.*

With the increasing use of alternating current in shipboard installations, it behooves every marine engineer to have as clear a picture as possible of a few of the basic operations accompanying its use. Of these, the paralleling of alternating current generators has always been extremely puzzling to the average engineer, accustomed to the simple single switching of D.C. shunt machines, and the almost equally positive manipulation of D.C. compound machines. Alternating current generators go into a new dimension of complication; and fluttering lights, or swinging indicator needles do not

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adequately explain the difference between success and failure in the "pull in." So, with a series of relatively simple vector analyses, and a painstaking effort to clarify each step as he progresses, the author in this article covers the possibilities and the difficulties of the paralleling operation in an effort to bring out the reasons for the simple rules which must be followed.

The purpose of paralleling generators is to divide the total load between the various units installed in proportion to the capacities of the machines. The accomplishment of load division by paralleling affords the designer an economical choice in the number and ratings of generators to be installed. The theory of parallel operation of direct current generators, particularly shunt wound machines, is relatively simple and well known to the personnel of the Navy. The theory of parallel operation of alternating current generators is, however, not as simple since there are a number of inherent factors which affect the operation of alternators that are not present in direct current machines.

The theory of parallel operation of electrical generators has been thoroughly and ably covered in many electrical text books and technical papers. The writer believes that in view of the adoption of alternating current for ship's service power and lighting on surface ships commencing with the *Farragut* it is worth while at this time to briefly point out and discuss from the theoretical standpoint the inherent differences between the parallel operation of direct and alternating current generators.

In the preparation of this paper the writer found the following books especially helpful:

Electrical Circuits and Machinery; Vol. I, Continuous Currents—Morecroft and Hehre.

Electrical Circuits and Machinery; Vol. II, Alternating Currents—Morecroft and Hehre.

Alternating Current Electricity, second course—"Timbie and Higbie."

Principles of Alternating Current Machinery—"Lawrence."

Prime mover control, Woodward Governor Company, Vol. VIII, February, 1937.

DIRECT CURRENT GENERATORS—SHUNT.

The direct current generators installed on surface combat type ships are turbine driven and vary in capacity from 25 Kw. on destroyers to 750 Kw. units installed on the large airplane carriers. The generators on the more recent ships are of the shunt wound type and have a drooping voltage characteristic. That is, when operating at rated speed and with the field rheostat in the same position, the voltage at rated current is about 12 per cent less than the no-load voltage. The purpose of this drooping characteristic is to aid in paralleling and provide smoother operation.

DIVISION OF LOAD BETWEEN SHUNT GENERATORS OPERATING IN PARALLEL.

The terminal voltage of a shunt generator is equal to the generated voltage minus the IR drop in the armature. The generated voltage is a function of the current flowing in the shunt field. Figure 1 shows two shunt generators connected in parallel.

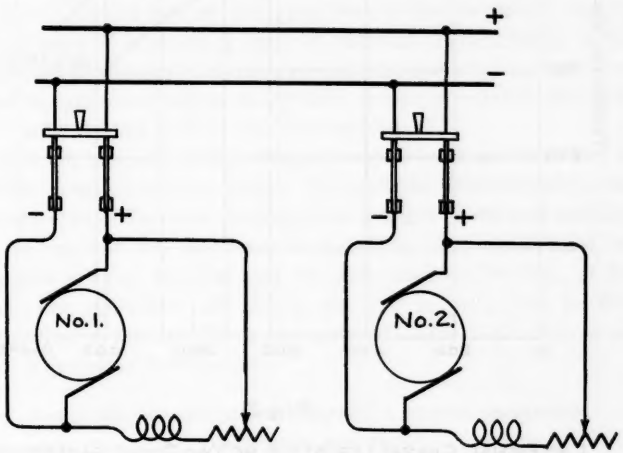

Fig. 1.

TWO SHUNT GENERATORS CONNECTED IN PARALLEL.

Since the two generators are in parallel the terminal voltages of the two machines must be the same since they are connected to the same bus-bars. If the two generators are of the same rating, have identical electrical characteristics and operate at the same speed, they will divide the load equally if the current flowing in the shunt fields is the same. The division of load between the two machines can be regulated by varying the shunt field current by means of the field rheostats. This phenomenon can readily be explained by means of Figure 2, which shows the external characteristics of two shunt generators running at constant rated speed.

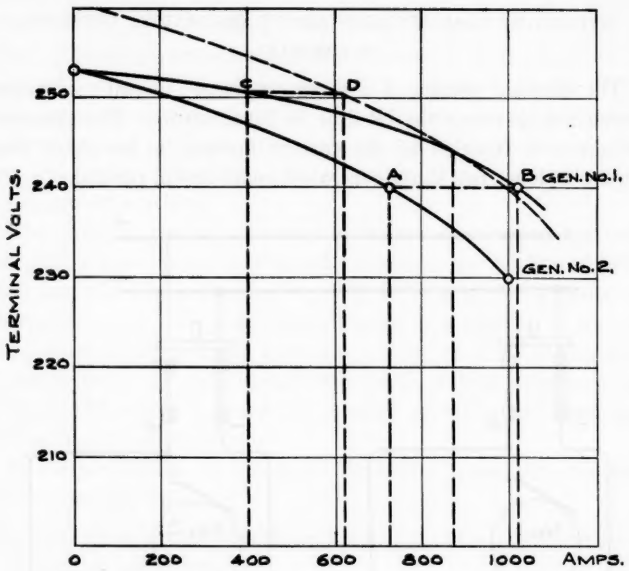

Fig. 2.

EXTERNAL CHARACTERISTICS OF TWO SHUNT GENERATORS.

Referring to Figure 2, it will be noted that with a total load of 1750 amperes, No. 1 generator is supplying 1020 amperes (point B), and No. 2 generator 730 amperes (point A). The equalization

of load between the two generators can be accomplished by increasing the field current in No. 2 generator, or by decreasing the field current in No. 1 generator. If the total load of 1750 amperes, at 240 volts, is to remain fixed and the load is to be divided equally then the field current in No. 2 generator must be increased and the field current in No. 1 generator must be decreased. Suppose that the field current of No. 2 generator is increased so that the load between the two machines is equally divided. Due to the increase in field current the external characteristic of No. 2 generator is indicated by the dotted line. The new external characteristic will be of the same shape as the previous one, but will be raised over the entire range of the curve. The load on the two machines will be equalized at 875 amperes, the point at which the two curves cross. At lighter loads No. 2 generator will take more current than No. 1 generator; for example, when No. 1 generator supplies 400 amperes (point C), No. 2 generator will furnish 620 amperes (point D).

If it is desired that the two generators divide the current equally at all loads it is essential that the external characteristics of the machines be identical. If the external characteristics are not identical the generator having the greater drooping external characteristic will take less than its share of the load.

It is not necessary in operating generators in parallel that they be of the same capacity or rating. The external characteristics must, however, be of the same shape and the voltages of the two machines at no load and full load must be the same. For example, if one machine is rated 250 Kw. and the other machine 500 Kw., at 240 volts, the generators will divide the load in proportion to their capacities; in this case No. 1 generator will carry 1040 amperes and No. 2 generator 2080 amperes.

PARALLELING AND DISCONNECTING A SHUNT GENERATOR FROM THE BUS-BARS.

The actual operations involved in paralleling shunt generators are simple and easily accomplished. To parallel a shunt generator with one already connected to the bus-bars, the incoming machine is brought up to rated speed and its terminal voltage adjusted equal to

the bus bar voltage and the generator switch closed. Under these conditions the incoming machine will take no current because the generated voltage is the same as the terminal or bus-bar voltage. However, by increasing the shunt field current the generator can be made to take the value desired. Increasing the shunt field current increases the generated voltage above the bus-bar voltage and this difference in voltage causes current to flow in the generator. Conversely in disconnecting a shunt generator operating in parallel from the bus-bars, the field current is decreased until the machine is carrying no current and the generator switch opened.

COMPOUND WOUND GENERATORS.

A compound wound generator differs from a shunt generator in that the former has a series field, and the current flowing in the armature flows through the series field coils. Since the series field coils are wound on the same cores as the shunt field coils the effect of the current passing through the series field coils in cumulative wound machines is to augment or increase the strength of the shunt field in direct proportion to the armature current. In order to have smooth operation with flat or overcompounded generators in parallel, it is necessary to connect the series fields with a conductor of very low resistance termed an "equalizer." Without the equalizer the operation of flat or overcompounded generators in parallel is very unstable in that they will not divide the load equally. In fact one generator will tend to take all of the load and run the other generator as a motor.

DIVISION OF LOAD BETWEEN COMPOUND GENERATORS RUNNING IN PARALLEL.

Figure 3 shows the external characteristics of two overcompounded generators. The generators have the same terminal voltages at no load, but are different at rated load. The generators have a 3 and 5 per cent rise in voltage, respectively.

Let us assume that the total load on both generators is 960 amperes, generator No. 1 will carry 600 amperes (point A) and generator No. 2 will carry 360 amperes (point B). The terminal voltage of the machines at this load is 245 volts. Suppose for some reason, the load on No. 2 generator is increased to 440 amperes

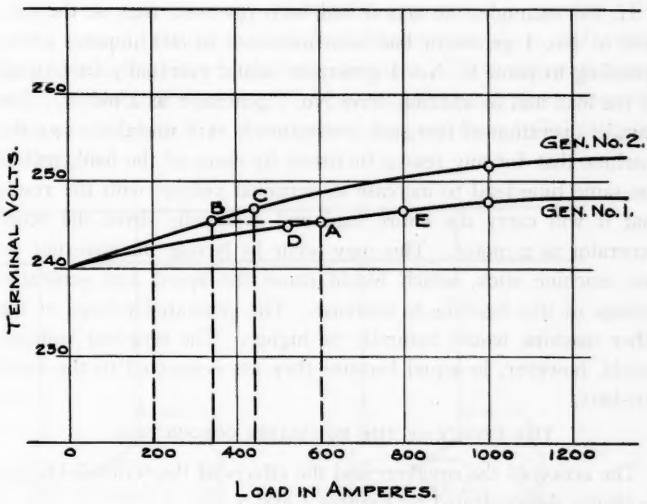

Fig. 3.

EXTERNAL CHARACTERISTICS OF TWO OVER COMPOUNDED GENERATORS.

corresponding to point C. Since the total load is 960 amperes, No. 1 generator would be expected to carry 520 amperes corresponding to point D. However, since No. 2 generator is operating at point C on the curve, its generated voltage has increased and is higher than the generated voltage of No. 1 generator corresponding to point D. The terminal voltages are actually the same since they are connected to the same bus-bars. The increased voltage generated in the windings of No. 2 generator will cause more current to flow in this generator. The increased current flowing in the series field will further increase the voltage of No. 2 generator. The decrease in voltage generated in the windings of No. 1 generator will cause less current to flow in this generator. The decreased current flowing in the series field will further decrease the voltage of No. 1 generator. This action continues until the series field of No. 1 generator reverses and it runs as a motor delivering power to its prime mover. Under these conditions No. 2 generator would not only supply the load of 960 amperes, but also the load of No. 1 generator running as a motor.

If, for example, the action had been reversed, that is, the current of No. 1 generator had been increased to 800 amperes corresponding to point E, No. 1 generator would eventually furnish all of the load and in addition drive No. 2 generator as a motor. The parallel operation of two such generators is very unstable, since the machine that for any reason increases its share of the load, will at the same time tend to increase its terminal voltage with the result that it will carry the entire load and eventually drive the other generator as a motor. This may occur by having the governor on one machine stick, which would cause the speed and generated voltage of this machine to decrease. The generated voltage of the other machine would naturally be higher. The terminal voltages would, however, be equal because they are connected to the same bus-bars.

THE EFFECT OF THE EQUALIZER CONNECTION.

The action of the equalizer and the effects of the series fields can be clearly demonstrated by Figures 4 and 5.

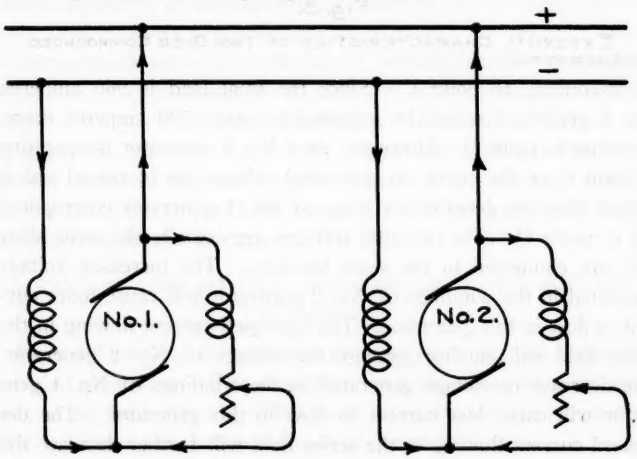

Fig. 4.

TWO COMPOUND GENERATORS CONNECTED TO BUS-BARS WITHOUT THE EQUALIZER CONNECTION-SHOWING THE DIRECTION OF THE FLOW OF THE CURRENTS.

Figure 4 represents two compound generators in parallel. It will be noticed that the current flows through the shunt and series fields of both machines in the same direction. If the electrical characteristics and the speeds of the two machines are the same, the terminal voltages will be the same and the generators will divide the load equally.

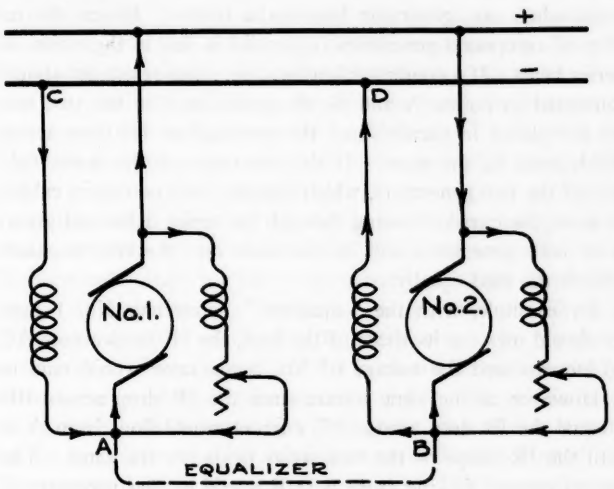

Fig. 5.

TWO COMPOUND GENERATORS CONNECTED TO BUS-BARS SHOWING THE DIRECTION OF THE FLOW OF CURRENTS WHEN ONE GENERATOR TAKES ALL OF THE LOAD AND DRIVES THE OTHER AS A MOTOR.

In Figure 5 are shown the two compound generators with No. 1 machine taking the entire load and driving No. 2 as a motor. It will be noted that during this condition the currents in the shunt fields of the two generators flow in the same direction; while the currents in the series fields flow in opposite directions. In the case of No. 1 generator the current flowing in the series field tends to strengthen the excitation of the shunt field and raise the generated voltage. In No. 2 generator the current in the series field tends to weaken the excitation of the shunt field and reduce the generated

voltage, allowing still more current to flow through the armature of No. 2 machine. This condition will continue until the resultant field of No. 2 generator is small and the armature is then practically short circuited across the terminals of No. 1 machine.

It will be seen that while the direction of the current in the shunt fields remains the same, the direction in the series fields is reversed when one generator becomes a motor. Hence the instability of compound generators in parallel is due to the action of the series fields. If a conductor having a very low resistance should be connected to points A and B, the series field of the two machines are placed in parallel, and the potential or IR drop across the fields must be the same. If the resistance of the series field circuits of the two generators, which includes the connecting cables, is the same, the current flowing through the series fields and armatures of both generators will be the same and the two machines will divide the load equally.

If, for example, with the "equalizer" connected, No. 1 generator should increase its share of the load, the IR drop across AC would increase and the voltage of No. 1 generator would tend to rise. However, at the same instant since the IR drop across BD must equal the IR drop across AC, current would flow from A to B, until the IR drops of the two series fields are the same. The amount of current flowing from A to B would be that necessary to equalize the IR drops, hence the term "equalizer."

The capacities of compound generators operating in parallel need not be the same. However, if machines of different capacities are to operate well in parallel and divide the load in proportion to the ratings of the machines, their no load and full load terminal voltages must be the same and the resistances of the two series field circuits must be inversely proportional to their ratings. For example, if one generator has a capacity one-half that of the other generator, the resistance of the series field circuit of the smaller machine should be twice that of the larger machine so that at all loads the IR drops of their series field circuits will be the same.

ALTERNATING CURRENT GENERATORS.

The essential difference between a direct and alternating current generator is that the former has a commutator. The function of

the commutator is to convert the generated alternating current into unidirectional or direct current. In the case of direct current, the instantaneous values are equal and the current is always in phase with the voltage. In the case of alternating current, the voltage and current are in phase only when the load has unity power factor, as for example, incandescent lamps or electric heating loads. Since the greater part of the load on a power station or on board ship consists of induction motors the current in practically every case lags behind the voltage. Since the current delivered by an alternator is not unidirectional, but is sinusoidal in shape, the instantaneous values vary from zero to maximum. (See curve E_1 Figure 6 (b).)

In this discussion the terms alternator and alternating current generator will be used indiscriminately. In a direct current generator the factors affecting the voltage as the load changes are the armature resistance, speed, field excitation and armature reaction. The terminal voltage is equal to the generated voltage minus the IR drop in the armature. In the case of an alternator the IX drop is an additional factor which must be considered. The reactance of an armature in ohms is equal to 2π times the frequency, times the self-inductance of the conductors ($2\pi f L$). The self-inductance of the conductors is caused by the magnetic lines of flux surrounding the conductors as a result of the current flowing through them. The current flowing in a reactance gives an IX drop expressed in volts. The reactance drop is displaced 90 degrees from the resistance drop.

The resistance of the armature of an alternator is not the same as the ohmic resistance measured by a wheatstone bridge. The alternating current flowing in the conductors produce a magnetic field around the conductors in the iron. This causes hysteresis and eddy current losses in the iron and eddy current losses in the conductors. These losses are included with the straight ohmic resistance and called the effective resistance of the armature. The effective resistance multiplied by the current squared (I^2R) gives the total heat loss in the armature. The vector sum of the effective resistance and the reactance is termed the impedance (Z) of the armature. The terminal voltage of an alternating current generator is the generated voltage minus the impedance drop, or

$E_T = E_G - IZ = E_G - I\sqrt{R^2 + X^2}$

E_T = terminal voltage
E_G = generated voltage
Z = impedance of the armature
R = effective resistance of the armature
X = Reactance of the armature
I = current

PARALLELING OF ALTERNATING CURRENT GENERATORS.

To parallel direct current generators it was only necessary to bring the voltages to the same value and to insure that the positive and negative terminals of the machines were properly connected. Furthermore, it was found that the load distribution could be varied by increasing or decreasing the current flowing in the shunt fields.

In the case of alternators, before paralleling can be accomplished the machines must first be "synchronized." In order to accomplish this operation the following conditions must be fulfilled:

1. The terminal voltages of the alternators must be the same. Since the voltage waves are sinusoidal in shape the instantaneous values must be equal.
2. The frequencies of the alternators must be the same.
3. The voltages of the alternators must be in phase.
4. In the case of multi-phase machines the direction of phase rotation must be the same.
5. The voltage wave forms must be the same. The voltage wave form of an alternator is determined by the design of the machine and cannot be changed by the operator. The voltage wave forms of present day designed alternators are sufficiently uniform and sinusoidal in shape so that no troubles from that source are usually encountered.

It will be noted that these requirements are similar to those for direct current machines. However, in order that the voltages of the alternating current generators are equal at all times it is necessary that they have the same frequency and be in the correct phase relationship—in "synchronism."

In paralleling direct current generators it was only necessary to have a voltmeter to indicate the voltage of the machines. In order to parallel alternators it is necessary to have additional instruments to indicate when the machines have the same frequency and correct phase relationship. Synchronizing lamps, or an instrument called a "synchroscope" are used for this purpose. The effect on the synchronizing lamps when the frequencies, voltages and phase relations vary offers an excellent means for presentation of the principles involved in the parallel operation of alternating current generators.

Figure 6 shows two three-phase, 120-volt alternators connected to the bus-bars, and synchronizing lamps connected across the incoming machine. Let us assume that No. 2 alternator is carrying

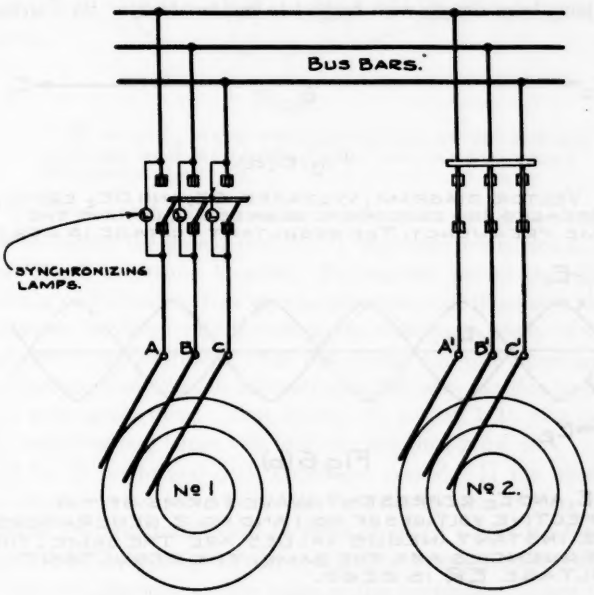

Fig. 6.

TWO ALTERNATING CURRENT GENERATORS CONNECTED TO BUS BARS SHOWING SYNCHRONIZING LAMPS ACROSS THE INCOMING ALTERNATOR.

the entire load and it is desired to parallel No. 1 alternator with the bus-bars. Number 1 alternator is brought up to rated speed and the field current adjusted so that the terminal voltage is the same as that of the bus-bars. The potential across the synchronizing lamps is the vector difference of the voltages of No. 1 alternator and the bus-bars when considered from the external load. However, when considered from the local circuit between the armatures of the two generators the potential across the synchronizing lamps is the vector sum of the generator voltages. If the frequencies and the voltages of the alternator and bus-bars have the same value and the generator voltage is in phase with the bus-bar voltage then the voltage difference will be zero and the lamps will remain dark. This condition is shown by Figures 6(a) and 6(b). These Figures have been drawn with *respect to the bus-bars or the external load.*

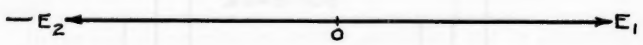

Fig. 6(a).

VECTOR DIAGRAM, VOLTAGES OE_1 AND OE_2 EQUAL, DISPLACED 180 ELECTRICAL DEGREES, AND HAVE THE SAME FREQUENCY. THE RESULTANT VOLTAGE IS ZERO.

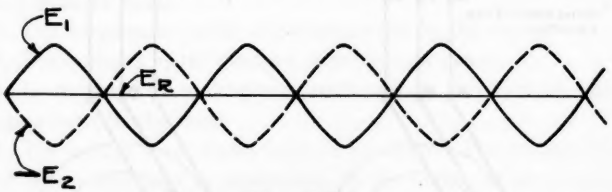

Fig. 6(b).

E_1 AND E_2 REPRESENT WAVE FORMS OF THE EFFECTIVE VOLTAGES OF NO. 1 AND NO. 2 GENERATORS. THE INSTANTANEOUS VALUES ARE THE SAME, THE FREQUENCIES ARE THE SAME. THE RESULTANT VOLTAGE ER IS ZERO.

CONDITIONS WHERE THE FREQUENCIES ARE DIFFERENT.

Let us suppose that the voltages of the two machines are the same as in the previous case, but that the frequency of No. 2 alternator is less than No. 1. This condition is represented by

Figure 7. E_1 represents the effective voltage of No. 1 alternator having a frequency of 60 cycles per second, E_2 represents the effective voltage of No. 2 alternator and its frequency is 50 cycles per second. During the same period as shown by Figure 7, that E_1 complete 6 cycles E_2 will complete 5 cycles. E_R is the resultant voltage and is the potential across the synchronizing lamps, which is equal to the difference of the voltages of the two generators $E_R = E_1 - E_2$.

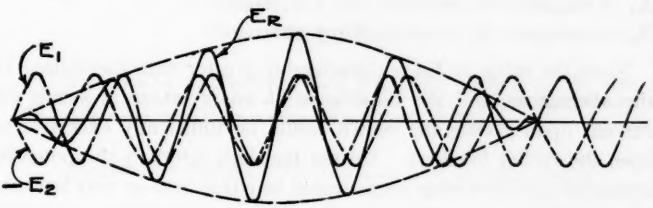

Fig. 7.

E_1 AND E_2 HAVE FREQUENCIES OF 60 AND 50 CYCLES PER SECOND. E_R IS THE RESULTANT VOLTAGE.

It will be noted that the resultant voltage E_R completes about 6½ cycles during the period that No. 1 alternator completes 6 cycles and No. 2 alternator 5 cycles. During this period the effective voltage of E_R varies from zero to maximum and then back to zero. It should be noted that the maximum voltage is nearly twice the voltage of one alternator. For this reason 120-volt lamps used to synchronize two 120-volt alternators would burn out due to having 240 volts across them. Thus during this period, 1/10 of a second, the synchronizing lamps will brighten and then grow dark. There will be 10 bright and dark cycles per second. If the alternator switch should be closed to the bus-bars during one of these bright periods the armature would be short circuited across the bus-bars and a local circulating current would flow in the armature circuits of the two alternators. The value of this circulating current would be the effective voltage of E_R divided by the sum of the impedances of the two armatures. Since the impedance of each armature is equal to the vector sum of the effective resistance and reactance

the value of the current may be expressed by the following formula:

$$I = \frac{E_R}{Z_1 + Z_2} = \frac{E_R}{\sqrt{(R_1^2 + X_1^2)} + \sqrt{(R_2^2 + X_2^2)}}$$

E_R = effective resultant voltage of E_1 and E_2
R_1 = effective resistance of armature No. 1 alternator
R_2 = effective resistance of armature No. 2 alternator
X_1 = reactance of armature No. 1 alternator
X_2 = reactance of armature No. 2 alternator

Since the value of E_R is considerably greater than the voltage of the alternators, and the impedance of an armature is small, the current upon closing the switch would be sufficiently excessive to open the circuit breakers. During the dark intervals the potential across the synchronizing lamps would be either zero or very low and no circulating current would flow upon closing the generator switch. The number of "light and dark periods" per second is equal to the difference of the frequencies of the two alternators.

CONDITIONS WHERE THE VOLTAGES ARE DIFFERENT.

Let us now study the effect on the synchronizing lamps when the frequencies are the same, the phase relationship is correct, but the voltage of one alternator is greater than the other. In the vector diagram, Figure 8(a), drawn with respect to the bus-bars, OE_1 represents the effective voltage of No. 1 alternator and OE_2 the effective voltage of No. 2 alternator. The resultant voltage is the voltage across the synchronizing lamps and is the vector difference of OE_1 and OE_2.

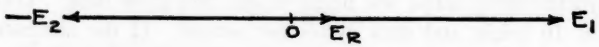

Fig. 8(a).

VECTOR DIAGRAM SHOWING OE_1 GREATER THAN OE_2. THE RESULTANT VOLTAGE IS OE_R.

As indicated by Figure 8(b), the resultant voltage E_R has the same frequency as E_1 and E_2. The value of E_R is less than that of E_1 or E_2, but its effective voltage may be large enough to cause the synchronizing lamps *to burn steadily.*

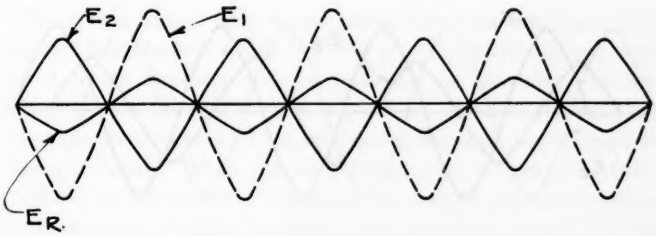

Fig. 8(b).

DIAGRAM SHOWING THE FREQUENCIES OF E_1 AND E_2 THE SAME. THE RESULTANT VOLTAGE E_R HAS THE SAME FREQUENCY AS E_1 AND E_2.

CONDITIONS WHERE THE VOLTAGES ARE OUT OF PHASE.

Let us assume the condition where the voltages and frequencies of the two alternators are the same, but the voltage of one machine leads the other by the angle $\emptyset$; that is the voltages of the two alternators are out of phase. This condition is shown by Figure 9(a) and 9(b), drawn with respect to the bus-bars or external load.

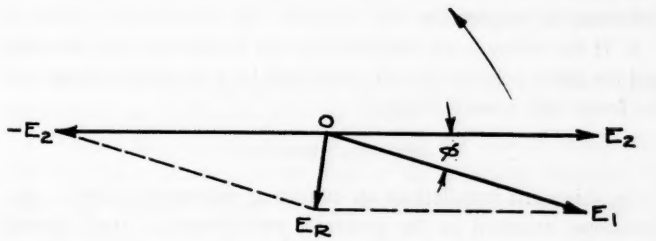

Fig. 9(a).

VECTOR DIAGRAM SHOWING OE_2 LEADING OE_1 BY THE ANGLE ϕ. OE_R IS THE RESULTANT VOLTAGE.

Figure 9(b) shows that the resultant voltage is sinusoidal in shape and has a constant effective voltage which will cause the *synchronizing lamps to burn steadily.*

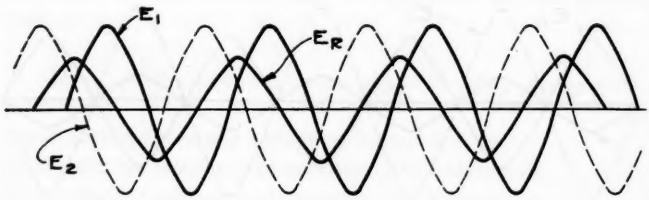

Fig. 9 (b).

E_2 LEADS E_1 BY 36 DEGREES, E_R IS THE RESULTANT VOLTAGE.

Summarizing, we find the effect on the synchronizing lamps during various conditions of voltage, frequency, and phase displacement is as follows:

1. If the voltages, frequencies and phase relations of the two alternators are the same, the resultant voltage is zero and the synchronizing lamps *will remain dark.*

2. If the voltages are the same but the frequencies are different, the synchronizing lamps will become *alternately bright and dark.* The number of bright and dark periods per second is equal to the difference in frequencies.

3. If the voltages are different but the frequencies are the same and the phase relation correct, there will be a resultant voltage and the *lamps will remain bright.*

THE SYNCHROSCOPE.

On shipboard installations an indicating instrument called a synchroscope, mounted on the generator switchboard, is used instead of lamps to indicate when the alternators are in synchronism. One type of instrument in use has a dial over which a pointer moves.

The instrument has a set of fixed coils and a movable coil. The fixed coils are connected to the bus-bars through a resistance while the movable coil is connected to the incoming machine through a condenser. When the pointer is in the middle of the scale the voltages of the incoming generator and the bus-bars are either in phase or 180 degrees out of phase. The scale on the dial is translucent and illuminated by a small synchronizing lamp.

The synchronizing lamp is connected across the secondary of a potential transformer which has two primary windings, one connected to the bus-bars and the other to the incoming machine. When the two voltages are 180 degrees out of phase the resultant voltage is zero, there is no potential across the lamp and the pointer cannot be seen. When the two voltages are in phase or exact synchronism the resultant voltage is the sum of the two voltages, the lamp burns brightly, and the switch should be closed. If the machines are out of synchronism the pointer will move to the right or left from the center of the dial depending on whether the speed of the incoming machine is slow or fast.

It was previously stated that the synchronizing lamps are *dark* when the generators are in synchronism. In the case of the type of synchroscope described above, it is necessary that the lamp be bright when the generators are in synchronism in order that the pointer can be seen. This is accomplished by proper connections of the transformer secondaries.

OPERATIONS NECESSARY IN PARALLELING ALTERNATORS.

To parellel an alternating current generator with one or more machines connected to the bus-bars the following operations are necessary:

1. Bring the incoming alternator up to approximately rated speed and connect the synchroscope in the circuit.

2. Adjust the voltage regulator so that the correct amount of field current is flowing in the field of the incoming generator to make the terminal voltage of the machine approximately that of the bus-bars.

3(*a*) Allow the incoming generator to speed up slightly faster than the synchronous or rated speed. The difference in frequencies must be very small, less than 10 cycles per second. When the synchroscope indicates that the machines are in synchronism the generator switch or circuit breaker should be closed.

(*b*) It is desirable that the incoming generator run slightly faster when connected to the bus-bars since the synchronizing action that pulls the machine into step will put a load on it and at the same time relieve the other machines on the line of some of their load.

If the incoming alternator is closed to the line while running slower than synchronous speed, it will take motor power to pull it into step.

4. As stated previously, to divide the load equally on direct current generators operating in parallel it is only necessary to adjust the field current flowing in the shunt fields. In the case of alternating current generators the load between the machines *cannot be adjusted by varying the field current.* To adjust the loads it is necessary to adjust the governors so that each prime mover delivers an equal amount of energy to its generator. This is accomplished on shipboard installations by governor pilot motors controlled from the generator switchboard.

SYNCHRONIZING CURRENT.

It has been shown that if at the instant of paralleling two alternating current generators the voltages are not in synchronism, or if the voltage of one alternator is higher than the other there will be a resultant voltage which will cause a circulating current to flow between the armatures of the two machines. The factors limiting the magnitude of this current are the resistance, inherent reactance, and the armature reaction. The effect of a leading current flowing in the armature is to strengthen the field and increase the induced voltage. Conversely, the effect of a lagging current is to demagnetize or weaken the field and reduce the induced voltage.

The circulating current, also frequently termed the " synchronizing current," is equal to the resultant effective voltage divided by the impedance of the two armatures: $I_R = \frac{E_R}{Z_1 + Z_2}$. Since the impedance limiting the synchronizing current is small a low resultant voltage will produce a large synchronizing current. This current produces heat in the armatures and reduces the useful load that the machines can carry.

Figure 10, drawn with respect to the *local circuit between the two armatures,* shows the condition at the instant of paralleling two alternators where the machines are in synchronism but the voltages are not the same. OE_R represents the resultant voltage and OI_R the synchronizing current. The synchronizing current will

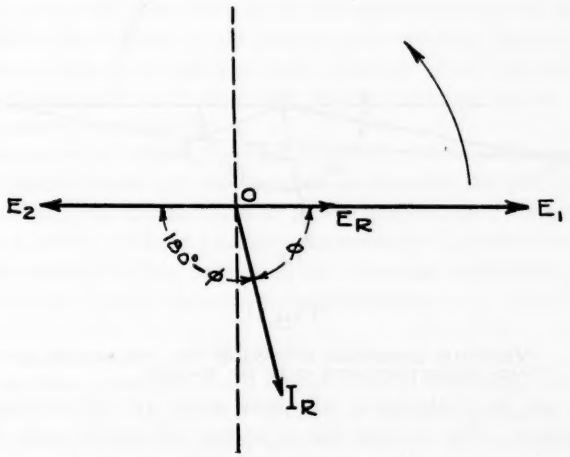

Fig. 10.

VECTOR DIAGRAM SHOWING THE VOLTAGE OF ONE ALTERNATOR GREATER THAN THE OTHER AND THE SYNCHRONIZING CURRENT.

lag behind the voltage of No. 1 alternator by the angle $\emptyset$, nearly 90 degrees, and will lead the voltage of No. 2 alternator *insofar as the local circuit is concerned,* by an angle equal to 180 degree — $\emptyset$. Since the synchronizing current OI_R lags OE_1 the effect will be to demagnetize the field of No. 1 alternator. At the same time, since the synchronizing current leads OE_2, the field of No. 2 machine will be strengthened. The resultant effect will be to bring the voltages of the two alternators closer together.

SYNCHRONIZING POWER.

If No. 2 alternator should be paralleled at the instant that its voltage was out of phase with No. 1 alternator as shown in Figure 11, there would be a circulating current OI_R flow between the two armatures. In order to avoid confusion, Figure 11 in this case has been drawn without an external load and viewed from the local circuit between the two generators. The machine that is leading, in

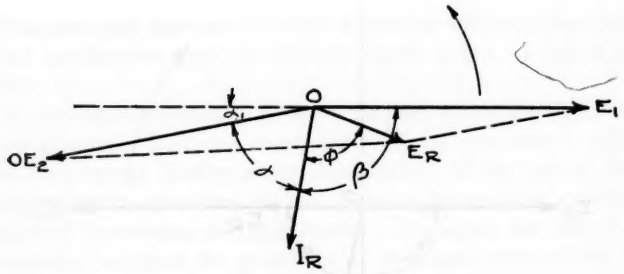

Fig. 11.

VECTOR DIAGRAM SHOWING THE VOLTAGES OF THE TWO GENERATORS OUT OF PHASE.

this case No. 2 alternator, will pump power into the machine that is lagging. The machine that is leading will tend to slow down, while the machine that is lagging will tend to speed up. Actually, since both alternators must run together at the same speed, the variation from synchronous speed will be only a matter of a few electrical degrees, sufficient to bring the machines back into the correct phase relationship. The power supplied by No. 2 alternator is $P_2 = \overline{OE_2} \times OI_R \times \text{Cos } \alpha$, and the power received by No. 1 alternator is $P_1 = \overline{OE_1} \times OI_R \times \text{Cos } \beta$.

THE POWER TENDING TO BRING THE TWO ALTERNATORS INTO PHASE IS TERMED THE SYNCHRONIZING POWER.

If the reactance of the armatures is large compared to the resistance the angle ϕ will be large and the angle α smaller. If the reactance is reduced the angle ϕ will be smaller and the angle α larger. Since the synchronizing power is greater when α is smaller, it is desirable that alternators be designed with a high value of armature reactance compared to the resistance in order *that the synchronizing power be large for small angles of phase displacement.*

HUNTING.

It has been shown that if two alternators operating in parallel "get out of step" the "Synchronizing Power" will tend to pull them back into step or synchronism. The leading machine supply-

ing the power will slow down, while the lagging machine will speed up. However, because of the inertia of the rotating masses, the generators will swing past the point of proper phase relationship. The synchronizing power will then reverse itself and tend to pull the machines back into step.

This period of hunting or oscillation is normally dampened by the hysteresis losses and eddy currents produced in the pole faces. Further damping can be obtained by the introduction of windings in the pole faces, termed "amortisseur windings." Turbine driven alternators require but little damping. Internal combustion and Diesel driven alternators require strong damping.

DIVISION OF LOAD.

It has been previously stated that the load of alternating current generators cannot be adjusted by varying the field excitation, and that to make an incoming generator take its share of the load it is necessary to admit more steam to the prime mover in the case of a turbine or more fuel in the case of a Diesel engine. This increases the power input to the alternator and the machine tends to speed up. Since the two generators are in synchronism the machine can only advance a few electrical degrees. However, in that small fraction of time it will pull ahead of the other generator and take more load.

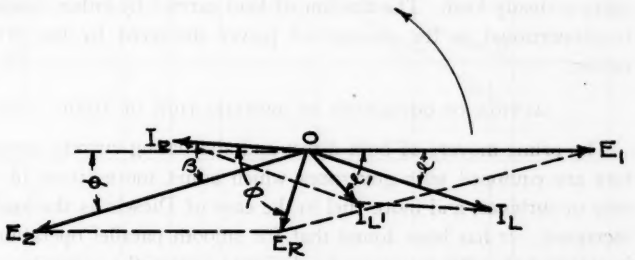

Fig. 12.

VECTOR DIAGRAM SHOWING THE INCOMING GENERATOR OUT OF PHASE BY THE ANGLE θ.

In Figure 12, which has been drawn with respect to the local circuit between the two armatures, OE_1 represents the terminal

voltage of generator No. 1, delivering a load OIL lagging by the angle α. The power delivered by No. 1 generator is $OE_1 \times OIL \times \text{Cos } \alpha$. Assume that generator No. 2 has pulled ahead by the angle θ due to an increase in steam to the turbine. Due to the difference in phase there will be produced a resultant voltage OER which will cause the synchronizing current OIR to flow in the circuit of the two armatures. This current lags OER by the angle $\emptyset$. Since No. 2 generator is leading with respect to OE_1 it will supply power to No. 1 generator equal to $OE_2 \times OIR \times \text{Cos } \beta$. The amount of current now supplied by No. 1 generator has changed to OIL^1 and the corresponding power supplied by generator No. 1 is $OE_1 \times OIL^1 \times \text{Cos } \alpha'$. Both the current and power supplied by No. 1 generator is reduced due to the flowing of the synchronizing current OIR. In this explanation Figure 12 is used to illustrate the reason for reduction in load on No. 1 generator when No. 2 generator has pulled ahead. Actually as soon as No. 2 generator has drawn ahead a very small amount due to admission of more steam to the prime mover, it would deliver some current to the external load in addition to the circulating current between the two armatures. The amount of current supplied by No. 2 generator to the external load would be the vector difference between IL and IL^1. As soon as No. 2 generator starts to supply power its speed is reduced and equilibrium is reached. At this point both alternators carry a steady load. The amount of load carried by either machine is proportional to the amount of power delivered by the prime mover.

ACTION OF GOVERNORS ON DISTRIBUTION OF LOAD.

The prime movers of both direct and alternating current generators are equipped with governors which admit more steam in the case of turbines, and more fuel in the case of Diesels as the load is increased. It has been found that for smooth parallel operation it is essential that the governors be designed so that the prime movers operate with from 3 to 6 per cent drop in speed between zero and full load.

It has been shown that in the case of direct current generators operating in parallel the load is adjusted or equalized between the two machines by increasing or decreasing the shunt field current.

The increase in load is directly proportional to the increase in the generated voltage. Upon increase of load the speed of the generator will slow down sufficiently to allow the governor to admit more steam or fuel oil to the prime mover as the case may be.

However, in the case of alternating current generators in parallel it is essential that the machines *all run at the same speed,* and as near constant speed as possible at all loads. In fact it has been shown that if for any reason one alternator runs at a different speed than the other, synchronizing power will be developed which will tend to pull the alternators back into step. This action aids the governors in keeping the speed of the two machines the same, and is not applicable to D.C. generators. Thus we see that for alternating current generators it is desirable that the prime movers all be equipped with governors having the same speed regulation.

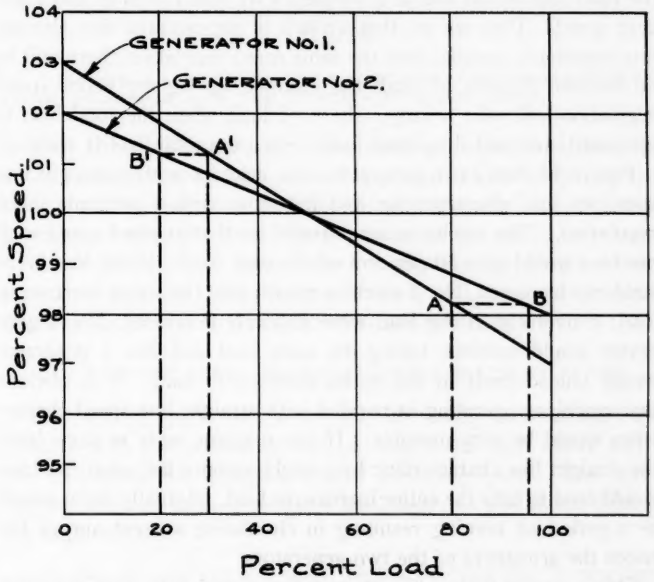

Fig. 13.

DROOPING SPEED-LOAD CHARACTERISTICS OF TWO GENERATORS.

The effect of speed regulation on load distribution is shown by referring to Figure 13. Generator No. 2 has a speed regulation of 4 per cent, while generator No. 1 has a speed regulation of 6 per cent. The governors are so adjusted that at 50 per cent load each generator is operating at 100 per cent or rated speed. Let us assume that while both generators are running at 50 per cent load, an additional load of 76 per cent is thrown on the line, generator No. 2 will take 46 per cent of the load (point B) while generator No. 1 will only take 30 per cent of the load. If the generators have been adjusted to operate at 60 cycles at rated load, both generators will operate at the same frequency, 59.5 cycles (98.2 per cent speed). Conversely, if the load were suddenly reduced by 50 per cent, generator No. 2 would drop 30 per cent of the load, while generator No. 1 would drop only 20 per cent of the load. Both of the machines would run at a frequency of 60.8 cycles or 101.2 per cent speed. Thus we see that while it is not essential that alternators running in parallel have the same speed regulation, there will be an unequal division of load, the machine having the better speed regulation will take a larger share of load when the total load is increased, and will drop load faster when the total load is reduced.

Figure 14 shows two generators, one machine with a straight line speed vs. load characteristic, and the other with 4 per cent speed regulation. The machines are adjusted so that at rated speed each machine would take 50 per cent of the total load. If the load were suddenly increased, No. 1 machine would take the entire increase in load; conversely, if the load were suddenly decreased, No. 2 generator would continue taking the same load and No. 1 generator would unload itself by the entire decrease in load. It is obvious that machines operating in parallel with straight line speed characistics would be very unstable. If one machine were to vary from the straight line characteristic by a slight amount the other machine would tend to take the entire increase in load. Actually there would be a period of hunting resulting in circulating current surges between the armatures of the two generators.

Ship's service turbo alternators are equipped with governor pilot motors for admitting more steam to the turbines. The control switch is located on the generator switchboard within easy reach of the operator.

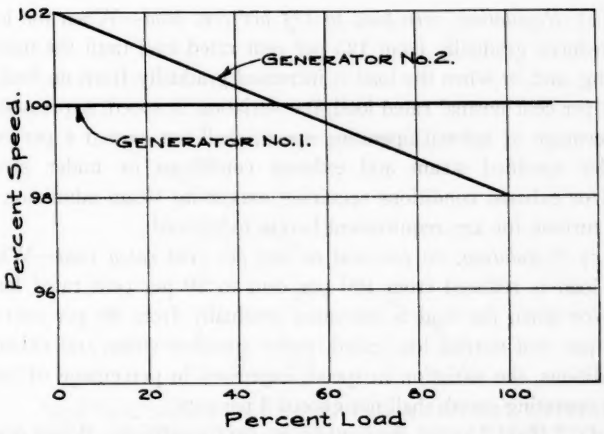

Fig. 14.

SPEED-LOAD CHARACTERISTICS OF TWO GENERATORS—ONE DROOPING AND THE OTHER A STRAIGHT LINE.

Since Diesel alternators are being installed on some of the ships under construction to take care of a portion of the ship's service load, it will be necessary that the Diesel engines be equipped with governors having the same characteristics as the turbo generators.

The importance of proper speed regulation for ship's service generators has been recognized by the Navy and the governor specifications for turbine driven generators are quoted below:

SHIP'S SERVICE TURBO GENERATOR GOVERNOR SPECIFICATIONS.

The rates of speed maintained shall be readily and manually adjustable with the turbine at rest, at least 5 per cent above and 5 per cent below the designed normal operating speed. The speed regulation of all turbo generators shall conform also to the following requirements:

(*a*) For all services the speed regulation, however momentary, shall not exceed 5 per cent of the designed normal operating speed of the turbine between the "no load" and "full load" conditions, when the load is applied or removed suddenly.

(*b*) *Regulation, zero load to 125 per cent load*—When the load is reduced gradually from 125 per cent rated load until the unit is idling, and/or when the load is increased gradually from no load to 125 per cent normal rated load, the variation in speed, expressed in percentage of normal operating speed, shall not exceed 4 per cent under specified steam and exhaust conditions or under steam and/or exhaust conditions requiring maximum steam admission to the turbine for any requirement herein mentioned.

(*c*) *Regulation, 20 per cent to 100 per cent rated load*—When the load is reduced from 100 per cent to 20 per cent rated load, and/or when the load is increased gradually from 20 per cent to 100 per cent normal load rated, under specified steam and exhaust conditions, the variation in speed, expressed in percentage of normal operating speed, shall not exceed 3 per cent.

(*d*) *"Hunt" speed fluctuation, no load condition*—When operating under the no load condition, periodic variations or fluctuations in speed shall not be in excess of 1 per cent from the normal operating speed.

(*e*) *"Hunt" speed fluctuation, load conditions*—For load variations not exceeding 10 per cent of the rated load and occurring at any part of its range between 20 per cent and 125 per cent rated load, the governor shall control the turbine speed within 1 per cent of the normal operating speed, even for momentary fluctuations.

(*f*) No adjustment of the governor, throttle valve, or connections shall be required or permitted to meet the foregoing requirements.

EFFECTS OF FIELD CURRENT VARIATION.

It has been repeatedly stated that variation in field current *has no effect on the distribution of load* of the alternators operating in parallel. We shall see that the effect of varying the field current is to cause a "synchronizing current" to flow in the local armature circuit *and to change the power factor* of the generators.

Figure 15, drawn with respect to the *bus-bars or external load*, represents two alternators in parallel, supplying a total load OI_L. The load supplied by No. 1 generator is OI_{L_1} and the load supplied by No. 2 generator is OI_{L_2}. The currents lag by the angle ϕ. Since

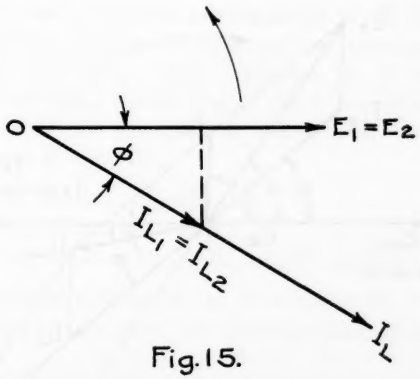

Fig. 15.

VECTOR DIAGRAM SHOWING CURRENT SUPPLIED BY EACH GENERATOR, WHEN THE VOLTAGES ARE THE SAME OI_L (LOAD CURRENT) = $OI_{L_1} + OI_{L_2}$.

the machines are in synchronism and assuming the speed regulation to be the same, OI_{L_1} and OI_{L_2} will have the same value, and their reactive components will be the same. The power supplied by the two machines may be expressed by the following formulae:

(*a*) $P_1 = OE_1 \times OI_{L_1} \times \text{Cos}\,\phi$
(*b*) $P_2 = OE_2 \times OI_{L_2} \times \text{Cos}\,\phi$
(*c*) Total Power $= P_1 + P_2 = OE_1 \times OI_L \times \text{Cos}\,\phi$

The wattless or reactive power which is useless power in that it is expended in heat and reduces the capacity of the machines is as follows:

(*d*) Reactive Power $= OE_1 \times OI_{L_1} \times \text{Sin}\,\phi$
(*e*) Reactive Power $= OE_2 \times OI_{L_2} \times \text{Sin}\,\phi$
(*f*) Total Reactive Power $= OE_1 \times OI_L \times \text{Sin}\,\phi$

Figure 16 has been drawn representing the voltage and current relationship with respect to the *local circuit between the two armatures of the two alternators operating in parallel* just after the field excitation of No. 2 generator has been increased. The generated voltage OE_2 is greater than the generated voltage OE_1 by the resultant voltage OE_R. The resultant voltage will cause the "synchronizing current" OI_R to flow, lagging OE_2 by the angle α,

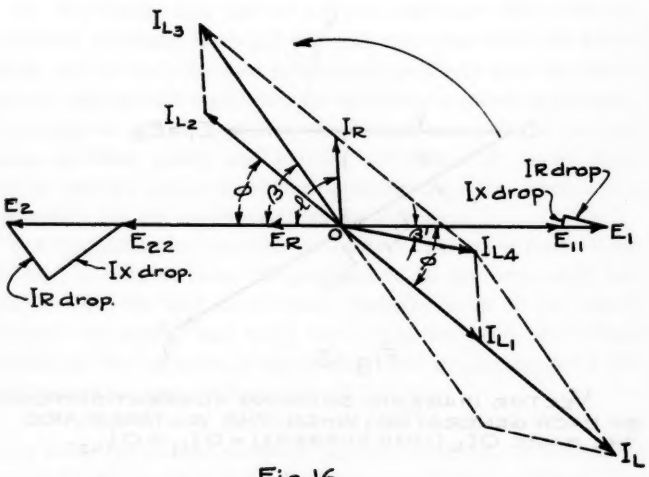

Fig. 16.

VECTOR DIAGRAM SHOWING TWO GENERATORS IN PARALLEL AT THE INSTANT WHEN THE VOLTAGE OF ONE IS GREATER THAN THE OTHER.

and leading OE_1 by the angle $(180° - \alpha)$. The actual current then flowing in No. 2 generator will be the vector sum of OI_{L_2} and OI_R or OI_{L_3}, while the actual current flowing in No. 1 generator is the vector sum of OI_{L_1} and OI_R or OI_{L_4}. The result has been to increase the current flowing in No. 2 generator and to decrease the current flowing in No. 1 generator, and at the same time to decrease the power factor of No. 2 generator and increase the power factor of No. 1 generator. No. 2 generator now carries most of the reactive current. It should be noted that the total load OI_L is still the same, and that the distribution of power between the two machines is about the same, since the power component of No. 2 generator $OI_{L_3} \cos \beta$ is practically the same as the power component of No. 1 generator $OI_{L_4} \cos \beta'$.

In fact the field current of No. 2 generator can be increased to the point where No. 2 generator is carrying all of the reactive current and the power factor of No. 1 generator is unity, or even leading.

Since the two generators are connected to the bus-bars, their terminal voltages must be the same and are represented on Figure 16 by OE_{11} and OE_{22}. E_1E_{11} represents the IZ drop in No. 1 generator, and E_2E_{22} the IZ drop in No. 2 generator. The power delivered by the two generators to the external load is expressed by the following formulae:

$P_1 = OE_{11} \times OI_{L_4} \times \text{Cos } \beta'$

$P_2 = OE_{22} \times OI_{L_3} \times \text{Cos } \beta$

Total Power $= P_1 + P_2 = OE_{11} \times OI_L \times \text{Cos } \phi$

It will be noted that the IR and IX drops of No. 2 generator are considerably greater than the corresponding drops in No. 1 generator.

The effect of increasing the field current of an alternator reduces its power factor and increases the power factor of the other machine. The reactive component of load is shifted to the machine with the increased field excitation. The power delivered to the external load by each generator does not change. However, by increasing the reactive component the I^2R loss in the machine is increased thereby decreasing the amount of useful load that the generator can carry.

TAKING AN ALTERNATOR OFF THE LINE.

In disconnecting an alternator from the bus-bars it is necessary to insure that no current is flowing in the armature circuit, otherwise the circuit breaker contacts may be burned, particularly if the voltage is high. Also with a constant load the sudden unloading of one generator would throw all of the load on the other generators and would cause an objectionable voltage dip. The first step in disconnecting the alternator from the bus-bars is to adjust the governor so that the power supply to the prime mover is reduced to the amount necessary to carry the no load losses and the wattmeter reads practically zero power. If there is any current indicated by the generator ammeters it will all be reactive or wattless current and this current should be reduced to zero by decreasing the field current. When both the wattmeter and ammeters read zero the circuit breaker should be opened, and the alternator disconnected from the line.

SUMMARY.

DIRECT CURRENT SHUNT GENERATORS.

1. Shunt generators of the same ratings operating in parallel will divide the current *equally at all loads* if the external characteristics (voltage—current curve, Figure 2) are identical.

2. If, however, the external characteristics are not identical the generator having the greater drooping characteristic will take less than its share of the load.

3. For successful parallel operation it is not necessary that shunt generators have the same rating. However, the shape of the external characteristics must be similar and the no load and full load voltages of the generators should be the same.

4. To parallel an incoming generator with those operating on the bus-bars the machine is brought up to rated speed, its terminal voltage adjusted equal to the bus-bar voltage and the generator switch closed.

5. The amount of current carried by a generator operating in parallel can be increased by increasing the current flowing in the shunt field.

COMPOUND WOUND GENERATORS.

6. When compound generators that do not have an appreciable drooping external characteristic are operated in parallel it is necessary to connect the series fields with a conductor of very low resistance termed an "equalizer."

7. Without an "equalizer," the operation would be unstable; one generator would tend to carry all of the load and eventually run the other machine as a motor.

8. For successful parallel operation it is not necessary that compound generators have the same rating. However, if compound generators are to operate well in parallel and divide the load in proportion to their ratings their no-load and full-load voltages must be the same and the resistances of the series field circuits must be inversely proportional to their ratings.

ALTERNATING CURRENT GENERATORS.

9. To parallel an alternating current generator with other machines operating on the bus-bars the following conditions must be fulfilled:

(*a*) The terminal voltage of the incoming generator must be the same as that of the bus-bars.

(*b*) The frequency of the machine must be the same as the frequency of the bus-bars.

(*c*) The voltage of the incoming generator in phase with the bus-bars.

(*d*) In multi-phase machines the phase rotation must be the same (taken care of when the machines are installed).

(*e*) The voltage wave form of the incoming generator must be the same as those connected to the bus-bars (taken care of in the design of the machines).

10. To parallel an incoming alternator with the bus-bars the following operations are necessary:

(*a*) Bring the incoming generator up to approximately rated speed and connect the synchroscope in the circuit.

(*b*) Adjust the voltage regulator so that the correct amount of field current is flowing in the alternator field to make the terminal voltage of the generator approximately the same as the bus-bars.

(*c*) Allow the incoming machine to speed up slightly faster than the synchronous or rated speed. The difference in the frequency should be very small, less than 10 cycles per second. When the synchroscope shows exact synchronism the circuit breaker should be closed.

11. The load between alternating current generators cannot be adjusted by varying the field current. To equalize or adjust the loads it is necessary to vary the output of the prime mover which is accomplished in the case of a turbine by varying the amount of steam admitted to the steam chest.

12. A circulating or synchronizing current will flow in the armatures of alternating current generators if one of the generators varies in phase relationship or if the voltage is higher than the bus-bar voltage. The circulating current heats the armatures and tends to reduce the capacities of the generators.

13. The synchronizing power tends to bring alternators back into step. It is very desirable that alternators be designed with a high value of armature reactance compared to the resistance in order that the synchronizing power be large for small angles of phase displacement.

14. For stable and smooth parallel operation of generators it is essential that the governors be designed so that the prime movers operate with from 3 to 6 per cent drop in speed between no load and full load.

15. It is not absolutely essential that alternating current generators operating in parallel have the same speed regulation. However, if the speed regulation is not the same the machine having the better speed regulation (less droop—Figure 13) will take a larger share of the load when the total load is increased and will unload itself faster when the total load is reduced.

16. Varying the field current of an alternator will have no effect on the load distribution. The effect of varying the field current will cause a " synchronizing current " to flow in the local armature circuit and change the power factors of the generators.

RESIDUAL STRESS.

By William C. Stewart, Member.*

In the November, 1936, issue of the Journal of the American Society of Naval Engineers, *Mr. H. W. Hiemke in his article, "The Repair of Steel Castings," referred briefly to the temperature-time relationships which exist when stresses must be released in a welded structure, and mentioned that a study of the matter was being made at the Naval Engineering Experiment Station, Annapolis, Md. In this article, Mr. Stewart, who conducted the study, discusses the findings at some length. It appears that it is impracticable to relieve all stress incident to a welding operation without destroying the basic strength of a structure, and that any stress relief is by way of being a compromise. In determining the proper temperature for stress relief, consideration must be given not only to the time of exposure, but to the metal being relieved, and the purpose for which the structure is intended. There is no simple rule for determining a stress relieving temperature.*

Stresses may exist in a body when the body is not subjected to any external forces or moments. Such stresses are referred to as internal, inherent, or residual stresses. Generally, these terms are used interchangeably in engineering practice.

Inherent stresses may be set up in a member by various means. During cold working of a bar the several layers are stretched in varying degrees so that each layer tends to take a definite length, the length of the bar being the equalization of the several lengths at some mean value. A similar phenomena occurs in wire drawing. On account of the draw plate restricting the outside of the wire, the interior layers in the longitudinal direction are stretched more than the exterior layers. This results in internal stress in the wire, and if the operation is carried too far, internal ruptures may occur.

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Hot-working or hot-forging of a bar that is not uniformly heated throughout will tend to increase the length of the outside layers over those of the inside, but since all are coupled together the several layers must have a common length. Obviously, excessive working of the bar may produce internal ruptures.

Another indication of inherent stress in metals is the so-called "season cracking." Severely worked brass, bronze and other materials may rupture after long periods of time. Failure in cold-worked materials may be induced as a result of fine scratches or the result of corrosion attack. Hard-worked brass is sensitive to several chemical reagents including ammonia fumes and solutions of mercury salts. The use of the latter for accelerating failure of severely cold-worked brass as a method of inspection is well known. The effects of cold-rolling on aluminum sheet is sometimes shown in tears and exfoliation under conditions where the material is subjected to corrosion attack.

Initial stresses are also present in unannealed glass. These stresses may produce double-refracting properties in the glass which are very undesirable from the standpoint of optical instruments.

In welded joints and structures inherent stresses result from non-uniform heating of the member. Stresses introduced in this manner are often referred to as "thermal stresses." Similarly, heating of a welded joint for the purpose of local stress-relieving will introduce other thermal stresses as a result of non-uniform heating. Conditions encountered in the casting of metals are somewhat analogous to those of welded joints. In fact, a weld consists of a relatively small casting of more or less refined structure fused to a chill mould. It is well known that inherent stresses are set up in castings as a result of variable section and the accompanying non-uniform temperature distribution. For this reason steel castings are placed in the annealing furnace immediately after shaking-out.

The presence of inherent stresses may not be revealed by any external signs; for example, suppose that a segment be cut from a steel ring and the ring deformed elastically, bringing the ends together. If the joint is welded and the weld surfaced so as to conform with the ring section, visual examination will give no indi-

cation as to the stress conditions of the ring. However, if the ring is cut radially, the ends will spring apart showing the presence of inherent stress.

When joining two steel plates by means of a butt weld, the plates are usually clamped in position on a heavy bedplate. This prevents warping of the assembly during welding. No distortion of the assembly will be encountered providing the plate is allowed to cool before removing the clamps. The stresses tending to warp the plate are relieved by plastic flow. Flow takes place most readily at high temperatures, hence changes in length resulting from temperature differences will be equalized by plastic flow. The permanently deformed regions cause stresses in the elastic vicinity where temperatures are not sufficiently high for plastic adjustment to occur. The presence of these stresses is not revealed by the appearance of the welded assembly.

Obviously, it is desirable that important welded structures such as high pressure boilers, turbine casings, steam piping and fittings, be as free as possible from residual stresses. The presence of residual stresses may contribute to the failure of the member and secondly, distortion of the assembly may occur after a time in service. Distortion of a turbine casing would be a serious matter since close tolerances are required for the satisfactory operation of the machine.

Residual stresses incurred by cold-working, welding and non-uniform heating can be practically eliminated by annealing. However, it is necessary that stress-relieving temperatures be selected giving due consideration to the type of alloy, and physical properties desired. Since metals and alloys are cold-worked for the purpose of increasing the mechanical properties, it is obvious that a compromise, between reduction in these properties and the extent of stress-relieving, must be accepted.

Stress-relieving temperatures for welded steel assembles should be as low as possible so as to avoid scaling and distortion. It is particularly important that complicated welded structures be adequately stress-relieved, without serious scaling or distortion. Moreover, it is necessary that the entire welded structure be heated and maintained at the stress-relieving temperature for a sufficient time to adequately stress-relieve the thickest section.

The necessity for information concerning the time-temperature relationship for stress-relieving welded structures resulted in the Naval Engineering Experiment Station undertaking a series of experiments for the purpose of establishing stress-relieving temperatures for steel and monel metal.

METHOD OF TEST.

In the experiments to be described, specimens having test lengths 0.505 inch in diameter and sufficiently long to accommodate 4-inch gauge marks were prepared of 0.27 per cent carbon boiler steel; boiler steel containing 0.21 per cent carbon and 1.44 per cent manganese; and hot rolled monel metal. Similar specimens were prepared of all-weld metal deposited with 3/16 inch diameter mild steel electrodes, Grade EA, Class 2. The specimens were threaded on both ends after the manner illustrated on Figure 1.

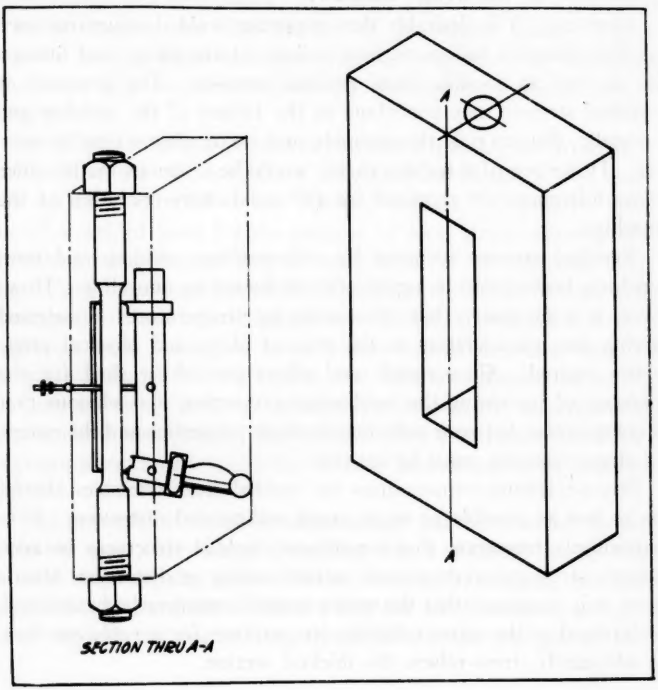


FIG. 1

The steel specimens were mounted in the heavy steel block shown in the sketch of Figure 1. The rectangular opening in the block permits the mounting of an extensometer on the test length of the specimen. Thus, with the extensometer mounted, the specimen can be stressed to definite values by tightening the nuts on the ends of the specimen. The heavy construction of the steel block makes the block practically non-deforming as compared with the test specimen. The specimen is strained to the yield point by tightening the end nuts. The magnitude of stress is indicated by the readings on the extensometer. The extensometer is removed from the specimen and the assembly, consisting of the strained specimen in the steel block, is placed in a muffle furnace. The assembly is heated at some predetermined temperature for a definite length of time, when it is removed from the furnace and allowed to cool. The extensometer is replaced on the specimen and the nuts released. Thus, the extensometer will indicate the magnitude of stress remaining in the specimen. Since the original and final conditions of stress are known, the difference represents the amount of stress relieved by reason of the heat-treating process. This procedure is repeated for different temperatures and different periods of time using a new specimen each time, in most cases.

A steel block is used for testing steel specimens in order to eliminate any effects that might result from differences in thermal expansion. Similarly, a monel metal block is used for testing monel metal specimens. The nuts are made of corrosion-resisting steel (18 per cent chromium, 8 per cent nickel) since this material does not scale appreciably at the temperatures used.

It is very important that the nuts be released without disturbing the extensometer. In order that only tensile stress may be applied to the specimen when the nuts are tightened, a keyway is cut in the end of the specimen provided with the movable nut. The specimen is prevented from twisting by keying it to the cross-member of the block. This precaution prevents any torsional stress being applied to the test length of the sample.

A word is necessary concerning the method of test described as applied to stress-relieving. As shown, the test consists of straining a specimen to the yield point and then determining the magnitude of stress remaining in the specimen following heat-treatment

at a definite temperature for a specified period of time. A similar condition exists when two plates are joined by a weld seam. If the edges of the plate are not restrained during welding the welding assembly will show a decided bow. If the edges are restrained a portion of the metal will have been strained to the yield point, otherwise the plate would not remain flat following removal of the clamps. The problem, then, is to determine the time-temperature relationship required to reduce the residual stress to a low value.

All-weld metal specimens representing three different conditions of deposition were also tested after the manner described. For one set of specimens the metal was deposited at such a rate that the temperature did not exceed 250 degrees F. for each pass, for the second lot the temperature did not exceed 320 degrees F. for each pass, and for the final lot the temperature did not exceed 400 degrees F. for each pass. Since the temperature of deposition for each pass has an effect on the stress set-up in the plate, it would be expected that the residual stress following stress-relieving temperatures would be different for different temperatures of deposition.

Since the steel and monel metal straining blocks are of much heavier section than the test specimens, it is essential that the temperature be brought up slowly so that the heavy section will be heated uniformly.

The extensometer used for all the tests is a type "A" Huggenberger tensometer. A ratio of approximately 1200:1 exists between the contacting knife edges of this instrument and the movement of the indicating pointer. Since one inch on the pointer scale can be sub-divided into about 80 parts, the instrument will show extensions of about .00001 inch per inch of test length.

RESULTS OF RELAXATION TESTS FOR 0.25 PER CENT CARBON BOILER STEEL.

Specimens of boiler steel having the following chemical and mechanical properties were tested at various temperatures and for various heating periods after the manner described in the previous section.

Boiler Steel.

Carbon, per cent	0.27
Sulphur, per cent	0.030
Phosphorus, per cent	0.006
Manganese, per cent	0.43
Silicon, per cent	0.01
Tensile Strength, pounds per square inch	63,600
Proof Stress, pounds per square inch	27,300
Elongation in 2 inches, per cent	35.8
Reduction of Area, per cent	55.8

The results of test are shown in the graphs of Figure 2. Residual stress is plotted as ordinates; tempering time as abscissas. Each graph represents tests at a constant temperature. It will be noted that tests at 800, 1000 and 1200 degrees F. were continued for 16 hours. The tempering effects observed represent conditions for a ½ inch diameter specimen in all cases.

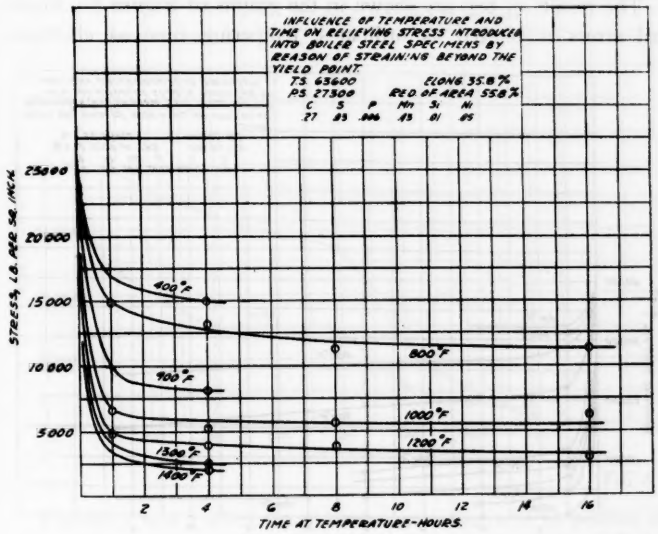


FIG. 2

RESULTS OF RELAXATION TESTS FOR BOILER STEEL CONTAINING 0.21 PER CENT CARBON AND 1.44 PER CENT MANGANESE

Specimens of manganese boiler steel having the following chemical and mechanical properties were tested at various temperatures and for various heating periods as previously described.

Manganese Boiler Steel.

Carbon, per cent	0.21
Sulphur, per cent	0.028
Phosphorus, per cent	0.007
Manganese, per cent	1.44
Silicon, per cent	0.01
Nickel, per cent	0.11
Tensile Strength, pounds per square inch	70,000
Proof Stress, pounds per square inch	34,900
Elongation in 2 inches, per cent	36.5
Reduction of Area, per cent	73.3

The results of test are shown in the graphs of Figure 3. Residual stress is plotted as ordinates, tempering time as abscissas.

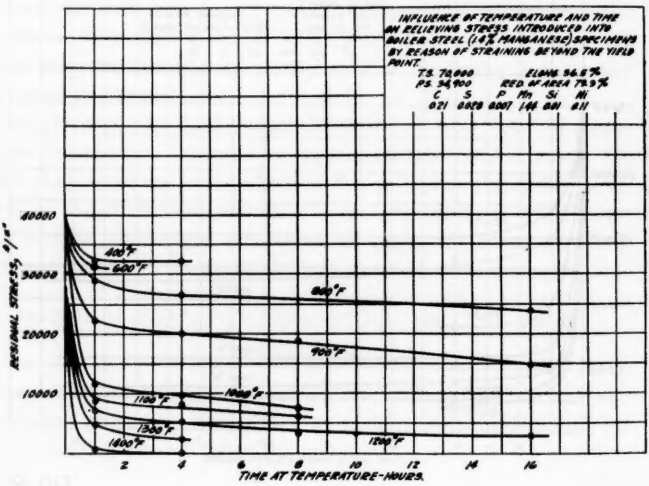


FIG. 3

Each graph represents tests at a constant temperature. The alternate use of the solid and open circle symbols has no significance except to avoid confusion in following the graphs. Tests at 800, 900 and 1200 degrees F. were continued for 16 hours. The tempering effects observed represent conditions for a ½ inch section.

Figure 4 presents graphs showing the stress-temperature relationship for the plain carbon and manganese boiler steels, designated materials KM and KL, respectively. The results shown represent tempering periods of four hours at the indicated temperatures. Both graphs show a gradual decrease in residual stress up to 800 degrees F. temperature. Above 800 degrees F. both graphs reveal a decided knee with rapid decrease in residual stress. Neither steel shows any considerable stress when held for four hours at 1200 degrees F. temperature. At a temperature of 1400 degrees F. there is practically no stress remaining in the samples.

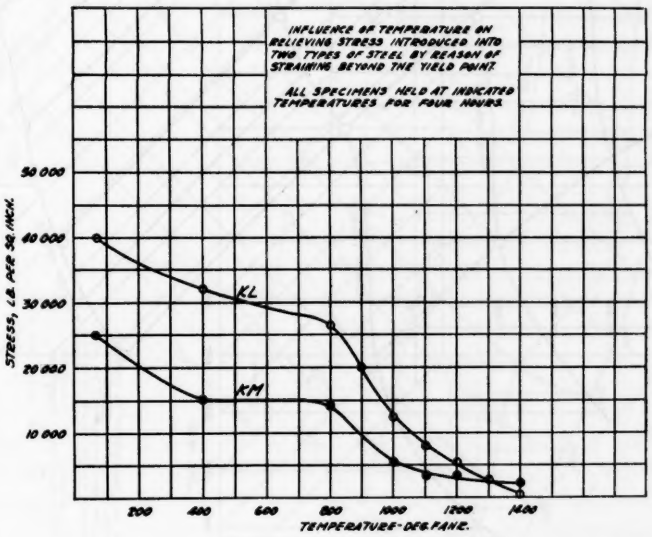
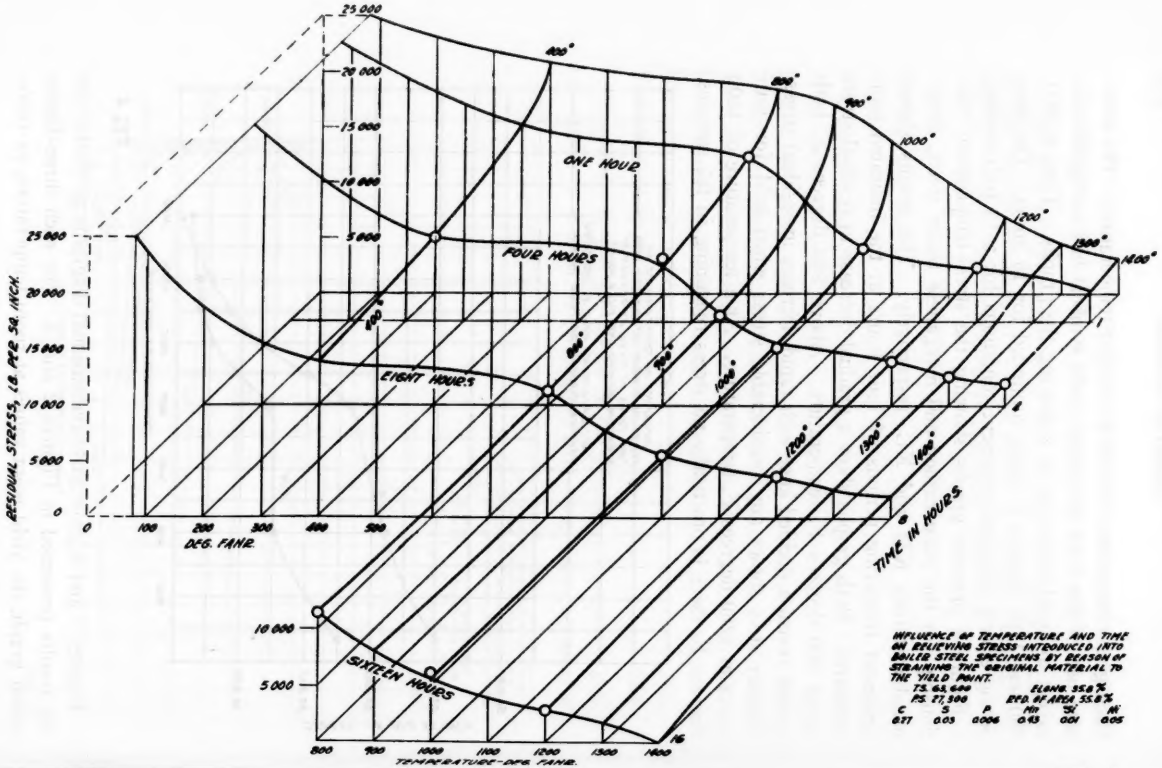


FIG. 4

Figures 7 and 8 show three-dimensional diagrams as plotted for the results presented in Figures 2 and 3. For each three-dimensional graph, the yield point values at room temperature, as repre-



INFLUENCE OF TEMPERATURE AND TIME ON RELIEVING STRESS INTRODUCED INTO ANNEALED STEEL SPECIMENS BY REASON OF STRAINING THE ORIGINAL MATERIAL TO THE YIELD POINT.

TS. 64,640 ELONG. 35.0 %
PS. 27,300 RED. OF AREA 55.8 %

C	S	P	Mn	Si	Ni
0.17	0.03	0.006	0.43	0.01	0.05

FIG. 7

FIG. 8

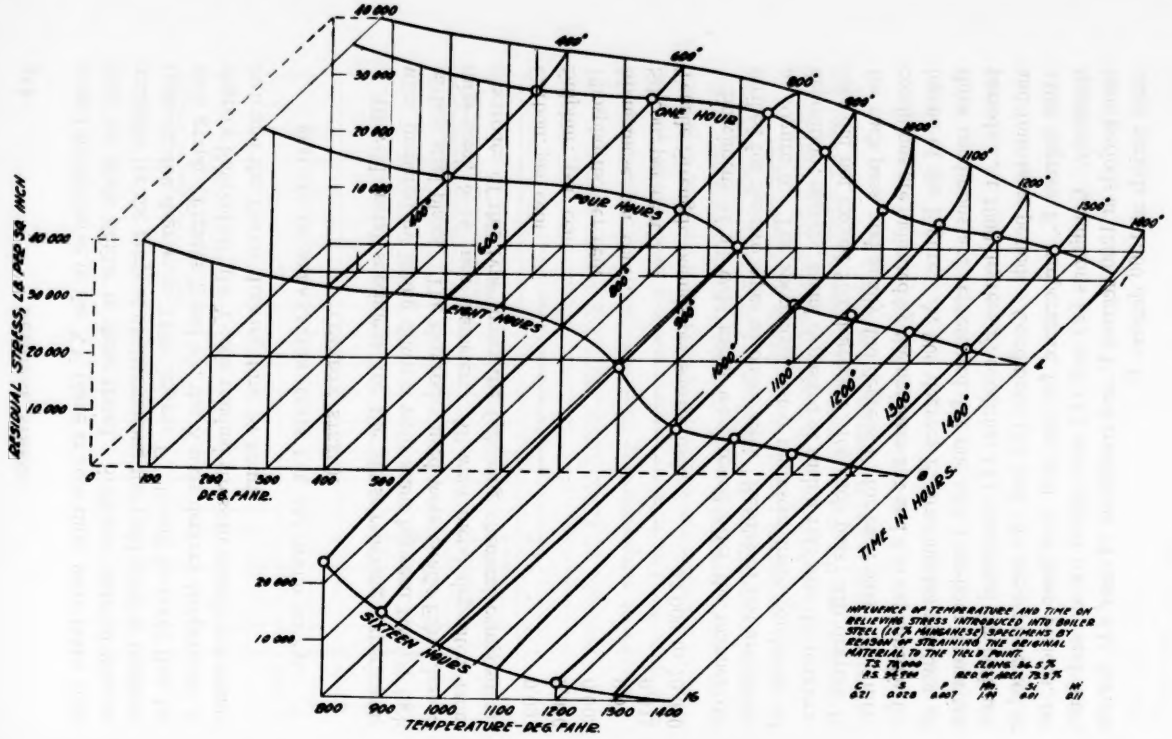

RESIDUAL STRESS, LB. PER SQ. INCH
40 000
30 000
20 000
10 000
0
100
200
300
400
500
DEG. FAHR.
ONE HOUR
FOUR HOURS
EIGHT HOURS
SIXTEEN HOURS
400°
600°
800°
900°
1000°
1100°
1200°
1300°
1400°
TIME IN HOURS.
800
900
1000
1100
1200
1300
1400
TEMPERATURE - DEG. FAHR.
INFLUENCE OF TEMPERATURE AND TIME ON RELIEVING STRESS INTRODUCED INTO BOILER STEEL (1.8 % MANGANESE) SPECIMENS BY MEANS OF STRAINING THE ORIGINAL MATERIAL TO THE YIELD POINT.
T.S. 79,000 ELONG. 26.5 %
R.S. 34,900 RED OF AREA 73.9 %
C .21 S .028 P .027 Mn .74 Si .201 Ni .11

FIG. 8

FIG. 7

sented by the curve in the XY plane at zero time, were taken from data for steels similar to those tested. Sufficient material was not available for the purpose of determining the yield point at temperatures of 400 degrees to 1400 degrees F. It will be noted that for each graph (Figures 7 and 8), the X-axis denotes temperature in degrees Fahrenheit; the Y-axis residual stress in pounds per square inch and the Z-axis drawing time in hours.

RESULTS OF RELAXATION TESTS FOR ALL-WELD METAL (CARBON STEEL).

All-weld metal specimens for the stress-relieving experiments were prepared by laying down a weld seam between two ¾ inch carbon steel plates. The electrodes used were Grade EA, Class 2, mild steel, 3/16 inch diameter. All-weld metal deposited from electrodes of this type shows the following chemical composition:

Carbon, per cent	0.09
Sulphur, per cent	0.010
Phosphorus, per cent	0.003
Manganese, per cent	0.55
Silicon, per cent	0.015
Tensile Strength, pounds per square inch	65,000 to 70,000

Specimens of all weld metal were tested after the manner described for the two types of boiler steel. Results of test are shown in Figure 5. The specimens tested represent three conditions of deposition; namely, metal deposited at such a rate that the temperature did not exceed 250 degrees F. for each pass; 320 degrees F. for each pass, and finally 400 degrees F. for each pass. The three conditions are indicated by the numbers 1, 2 and 3 as explained in the legend of the Figure. At 900 degrees F. drawing temperature, all three conditions are represented for one and four-hour tempering periods. At 1000 degrees F. condition (1) was tested for both one and four-hour periods. Conditions (1) and (3) are represented at 1100 degrees F. temperature for one and four-hour periods, respectively. Conditions (2) and (3) were tested for one and four-hour periods at 1200 degrees F. and conditions (3) and (1) for the same periods at 1300 degrees F.

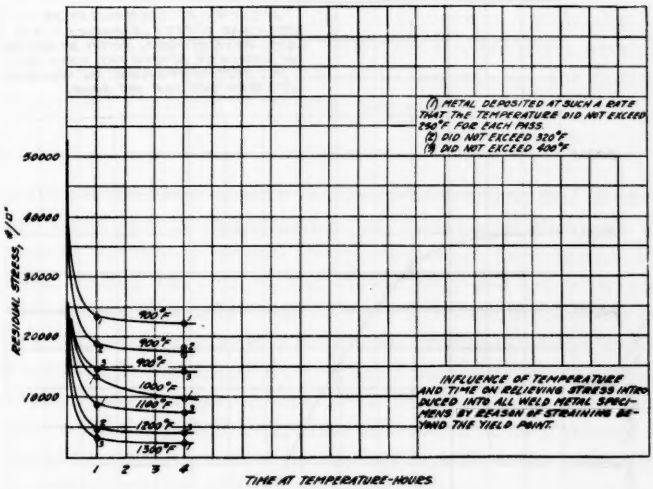


FIG. 5

RESULTS OF RELAXATION TESTS FOR MONEL METAL.

Results of stress-relieving tests for monel metal are presented in Figure 6. The tests were made using a monel metal block in order to eliminate the effects of differences in thermal expansion as previously mentioned. All specimens were maintained at the indicated temperatures for one hour. Consequently, the influence of various tempering periods is not shown for this material.

Hot rolled, 1 inch diameter monel metal rods were used in the preparation of the test specimens. The monel metal showed the following chemical and mechanical properties:

Chemical Composition.

Carbon, per cent	0.08
Sulphur, per cent	0.028
Phosphorus, per cent	0.009
Manganese, per cent	0.45
Silicon, per cent	0.03
Nickel, per cent	66.44
Copper, per cent	27.82
Iron, per cent	2.05
Aluminum, per cent	3.12

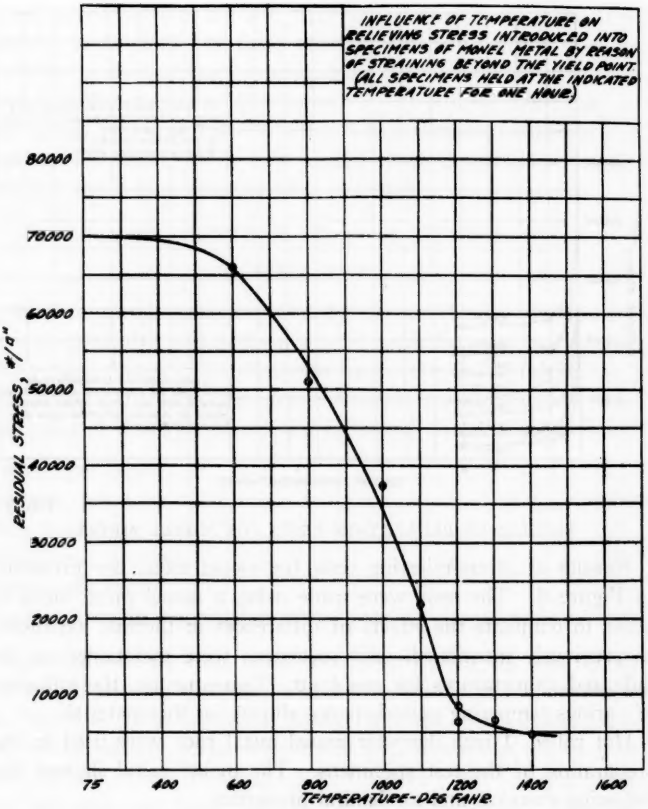


FIG. 6

Mechanical Properties.

Tensile Strength, pounds per square inch	74,500
Johnson's Limit, pounds per square inch	68,500
Proof Stress, pounds per square inch	71,500
Elongation in 2 inches, per cent	32.5
Reduction of area, per cent	57.6

The graph of Figure 6 shows that there is no considerable stress-relief at 600 degrees F. temperature for a ½ inch section after one

hour at temperature. At 800 degrees F. there is considerable decrease in residual stress. Low values for residual stress were obtained for temperatures of 1200 to 1400 degrees F. The material is completely softened at the latter temperature.

DISCUSSION.

An attempt has been made in the experiments described to establish a basis for the heat-treatment of welded assemblies with the view to removal of residual stresses occasioned by welding. The usual practice has been to heat carbon steel and low alloy steel welded assemblies to 1100-1200 degrees F. for at least one hour per inch of section. As far as is known, the practice of heating welded assemblies in this temperature range is based largely on practical experience. It appeared desirable to have available information concerning the probable magnitude of residual stresses relieved by various temperatures when the castings or sections are maintained at temperature for different periods of time.

The results of test presented for both the 0.27 per cent carbon boiler steel and the manganese boiler steel have shown that residual stresses occasioned by straining the material to the yield point remain in considerable amounts for stress-relieving temperatures below 1200 degrees F. These findings seem to conform to practical experience, since it is generally considered that a temperature of approximately 1200 degrees F. is necessary for complete stress relief.

From the results of test shown in Figure 4, it is evident that a stress-relieving temperature of 900 degrees F. for four hours, is effective in relieving a considerable proportion of the residual stress occasioned in the ½ inch test sections. For temperatures of 800 degrees F. and lower, stress relief is not considerable for the ½ inch sections, even after maintaining at temperature for 16 hours. The fact that a temperature of 900 degrees F. results in considerable stress relief is important, since it may be necessary to subject castings to stress-relieving temperatures after finish machining. Under these conditions the low temperature will result in much less warping of the casting than will a similar heat-treatment at 1200 degrees F.

The fact that the graphs of Figures 2 and 3 at 800 degrees F. show a slight downward slope indicates that residual stress in these steels is slowly relieved at this temperature. Accordingly, the claim by some that residual stress is relieved by creep after the welded assembly is placed in operation is justified in some measure for installations having relatively high metal temperatures. However, the evidence available indicates that a considerable part of the stress would not be relieved until after a relatively long time at 800 degrees F. Obviously, the higher the operating temperature, the more effective will be the stress relief. On the other hand, the higher the operating temperature, the more necessary it is that the assembly be in a stable condition as regards residual stress. Results of test obtained at 800 degrees F. do not indicate that residual stress will be reduced, even after a very long time, to a value comparable with the decrease obtained after one hour at 1200 degrees F.

The carbon steel shows approximately 5000 pounds per square inch residual stress after one hour at 1200 degrees F., approximately 4000 pounds per square inch after 4 hours and 3000 pounds per square inch after 16 hours at 1200 degrees F. A very similar condition is noted for the manganese boiler steel, Figure 3. It will be noted that the residual stress is not completely relieved even after one hour at 1200 degrees F. However, the remaining stress is sufficiently low so as not to affect the structure appreciably.

The experiments show that the temperature of stress-relieving is much more important than is the length of time at temperature. In fact, maintaining the ½ inch test samples at temperature for more than four hours has but little effect on residual stress. A temperature of 1200 degrees F. for one hour is more effective in relieving residual stress in a ½ inch section than is a temperature of 1000 degrees F. maintained for 16 hours.

Results of experiments for the all-weld metal specimens show that for relatively low stress-relieving temperatures; namely, 900 degrees F., the higher the temperature of depositing the metal, the more effective the stress-relieving treatment. Thus, the graphs of Figure 5 show that the residual stress in weld metal deposited at temperature not in excess of 400 degrees F. is considerably more relieved at a temperature of 900 degrees F. than is weld metal deposited so that the temperature of each pass does not exceed 250

and 320 degrees F. This behavior is not so noticeable for higher temperatures of stress-relieving for the reason that the residual stress is proportionately much less. At 1200 degrees F. temperature there appears to be little difference between the specimens deposited at temperatures not exceeding 320 degrees F. for each pass and specimens deposited at temperatures not exceeding 400 degrees F. for each pass.

The graph of Figure 6 shows the effect of temperature on the relief of residual stress in monel metal. No study was made of the effect of varying the time while at temperature; all specimens being maintained at temperature for one hour. The data presented may not be strictly applicable for severly cold worked material since the results were obtained with the hot-rolled alloy. However, welding is usually done on the latter type of material.

Stress relief is not pronounced at a temperature of 600 degrees F. for one hour for the ½-inch specimen. At 800 degrees F. there is a considerable decrease in residual stress. At 1200 degrees F. residual stress is reduced to a low value. Complete anneal and softening is encountered at about 1400 degrees F. For severely cold-worked material, a low temperature anneal at 800 degrees F. is usually adequate for relieving stress so that bar stock can be maintained true when machined. Where cold-worked monel metal is indicated for its high mechanical properties, a balance must be obtained between the amount of stress relief and the resulting physical properties. Obviously, stress relieving at 1200 degrees F. would affect markedly the physical properties of the cold-worked material. For this purpose a temperature of 800 degrees F. has been found satisfactory. Cold-worked monel metal, annealed at 600 degrees F., frequently shows erratic behavior when subjected to fatigue tests. The same material when annealed at 800 degrees F., shows normal behavior and stress-cycle graphs showing little scatter of the individual points can be obtained.

CALCULATION METHODS FOR POWER CIRCUITS USING ELECTRON TUBES.

By Rear Admiral S. M. Robinson, U. S. Navy, Member.*

This is the third and last of a series of articles which Admiral Robinson has contributed to the Journal *in successive numbers, covering the extension of electron tube development to power generation and transmission—a field so novel and revolutionary that the author properly suggests that it may well be called the "New Electricity." This article is of particular importance to the design engineer, and not to the random reader, for in it are derived the formulae upon which mutator machinery may be built and studied. While the development is only now emerging from a laboratory stage, it should be understood that a power transmission line embodying its principles has already been completed, and small electrical machines have been built to turn the theory into practicality.*

This is the third and final paper of a series dealing with electron tubes as used for power purposes. The first paper covered "power transmission," the second covered "alternating current motors used with electron tubes," and it is proposed, in this paper, to show how the calculations are made for the circuits shown in the first two papers.

At this point, it is desired to acknowledge, with thanks, the very generous assistance of Mr. A. H. Mittag, of the General Electric Company, without whose help these papers could not have been prepared at all; also that of Doctor E. F. W. Alexanderson, who is mainly responsible for the circuits shown.

The current and voltage calculations in these electron tube circuits are fully as different from the conventional calculations for

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alternating and direct current circuits as are the circuits themselves. For that reason, the formulae will be developed in rather great detail.

The first circuit to be considered will be that of the parallel circuit type of thyratron motor, described in the second paper mentioned above. This is shown in Figure 1. Here we have a motor

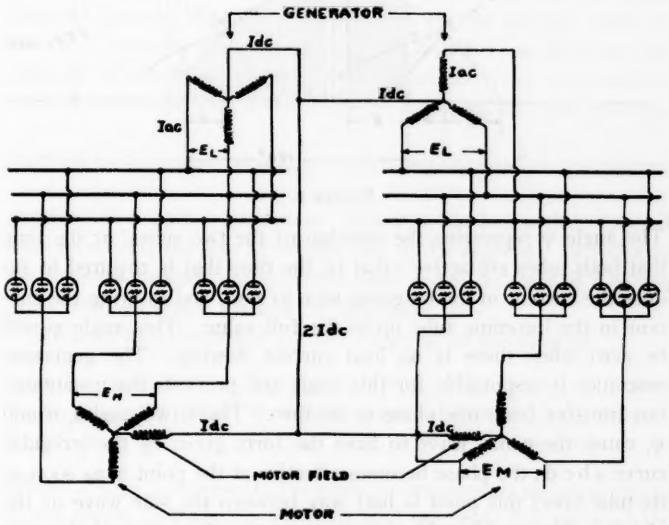

FIGURE 1.

with two windings on its armature, each connecting to the neutral of the two generator windings which are also on one armature. In other words, the figure represents one generator and one motor which are internally subdivided electrically. It is desired to find formulae which will give the currents and voltages in the different parts of the circuits shown.

To do so, we will begin with the generator itself. In Figure 2 is shown the e.m.f. wave of 1-phase of the generator—that is, the voltage as measured from generator neutral to generator terminal. In this figure, the angle α represents the amount of delay, or retardation (by the grid), of the firing of the tube which makes this

particular phase effective, as explained in paper No. 2, covering thyratron motors. If the tube were not retarded, it would fire at the intersection a′, of the sine waves of e.m.f. of two of the phases.

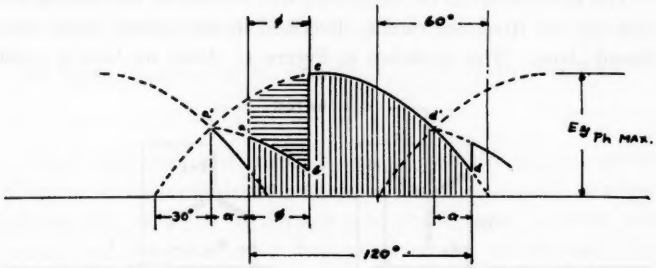

FIGURE 2.

The angle ψ represents the over-lap of the two tubes, or the time that both tubes are active—that is, the time that is required to reduce the current in the outgoing tube to zero, and to bring the current in the incoming tube up to the full value. This angle ψ will be zero when there is no load current flowing. The generator reactance is responsible for this angle and prevents the instantaneous transfer from one phase to another. These two angles, α and ψ, cause the e.m.f. wave to have the form given by the irregular curve a b c d; the phase becomes effective at the point a, as soon as its tube fires; this point is half way between the sine wave of the phase under consideration (a′ c d′) and the sine wave of the outgoing phase, which is 120 degrees from a′ c d′, as shown in the figure; the e.m.f. wave of the phase will follow a sine wave half way between the two sine waves, along the line a b as shown; when the current in the outgoing phase is reduced to zero, as shown at b, the e.m.f. curve will rise instantaneously to the point c on the sine wave of the phase we are considering; it will follow this curve till a tube is fired on another phase, as shown at d, when the process will be repeated. For the portion of the curve, covered by a b, we have two tubes passing current and these two tubes, plus the two phases of the generator, form a local short circuit, as will be seen by a reference to Figure 1. The difference in voltage between the phase a′ c d′ and the outgoing phase will be available for reducing the cur-

rent in the outgoing phase to zero and building that in the incoming phase up to load current; during this period, the total current will remain constant; it will be the sum of the currents in the two tubes and will be assumed to be maintained constant by the reactance of the motor field. If neither angle exists, that is if $\alpha = 0$ (no retardation) and the current is practically zero (so that ψ is also zero), then the e.m.f. curve will simply be that portion of the sine wave included between a′ and d′. In order to get the average value of the e.m.f. given by the curve a b c d, we shall have to find the area included between this curve and the base line (the area hatched with vertical lines) and divide it by the angle, on the base, which is included by this curve; this angle is 120 degrees, or $\frac{2\pi}{3}$, as shown, since this is a 3-phase generator. The area hatched with horizontal lines represents the loss of voltage due to the angle ψ. In order to get the area hatched with vertical lines, it will be best to find the areas included under a b and c d separately.

The area included under a b will be given by the integral

$$\int_{30^\circ + \alpha}^{30^\circ + \alpha + \psi} E_{G\ ph\ max}\ 1/2\ [\sin\theta + \sin(\theta + 120^\circ)]\ d\theta$$

Where $E_{G\ ph\ max}$ represents the peak, or maximum, value of e.m.f. in one phase, as shown in Figure 2, and the angle θ is the variable angle of the sine wave. The above equation follows from the fact that the line a b is a sine wave half way between the two sine waves given by the equations

$$E_{G\ ph\ max} \sin\theta = \text{e.m.f.}$$

and

$$E_{G\ ph\ max} \sin(\theta + 120^\circ) = \text{e.m.f.}$$

The ordinate of e.m.f. will therefore be half the sum of the two.

The area included under c d will be given by the integral

$$\int_{30^\circ + \alpha + \psi}^{150^\circ + \alpha} E_{G\ ph\ max} \sin\theta\ d\theta$$

If these two integrals are integrated and added together, we shall find the sum to be

$$\frac{\sqrt{3}}{2} E_{G\ ph\ max} [\cos (\alpha + \psi) + \cos \alpha]$$

Dividing this by $\frac{2\pi}{3}$ we have

$$E_{av} = \frac{\frac{\sqrt{3}}{2} E_{G\ ph\ max} [\cos (\alpha + \psi) + \cos \alpha]}{\frac{2\pi}{3}}$$

Where E_{av} = average value of the e.m.f. represented by the curve. Simplifying, we have

$$E_{av} = \frac{3\sqrt{3} E_{G\ ph\ max}}{4\pi} [\cos (\alpha + \psi) + \cos \alpha]$$

E_{av} is generally represented by E_{dc}, this being the equivalent direct current voltage, or that direct current voltage which, when multiplied by the direct current flowing in the motor field, will give the power of the motor.

$$\therefore E_{dc} = \frac{3\sqrt{3} E_{G\ ph\ max}}{4\pi} [\cos (\alpha + \psi) + \cos \alpha] \qquad (1)$$

The voltage $E_{G\ ph\ max}$ is not one that can be measured, since it is the internally generated voltage in each phase of the generator; we can only measure the voltage across the terminals of the generator and this will be an entirely different value since the RI and XI drop in the generator phase will both affect the value of the e.m.f. across the terminals. It is therefore necessary to obtain an equation which contains terminal voltage instead of generated voltage.

To obtain this equation it will be assumed that the two windings of the generator in Figure 1 are equally loaded and that the legs of the windings which are 180 degrees out of phase are coupled 100

per cent, so the two windings in one slot can be considered as one winding. With these assumptions it will be seen that a commutation taking place between two legs of one of the windings will have the same effect in the other winding as if the commutation had taken place in the latter winding. There will then be six commutation jogs per cycle in the voltage between any two terminals of either of the two windings. Only a half cycle of voltage need be considered here as the other half, in view of the above assumptions, is an exact duplicate of it; the half cycle will therefore contain only three jogs.

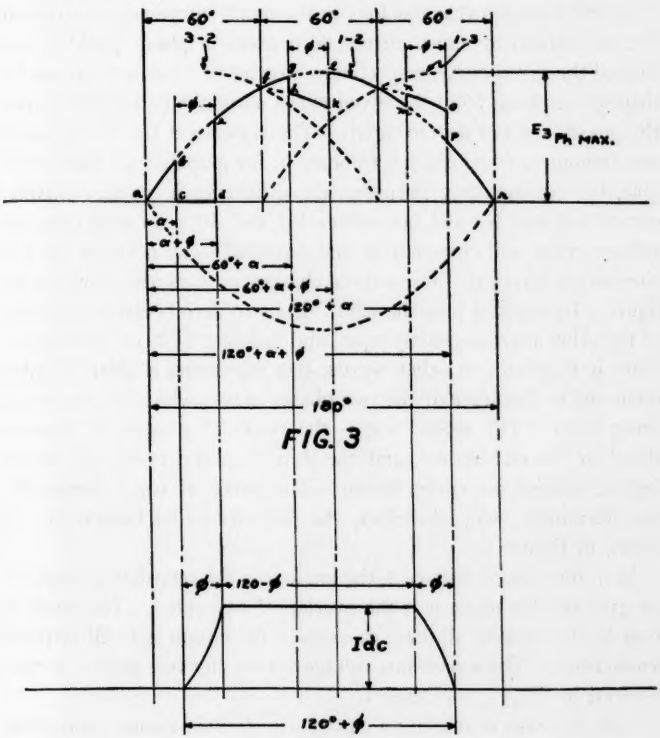

FIGURE 3-A.

Figure 3 shows one loop of the e.m.f. wave across any two of the generator terminals, of either of the two windings, or across any two lines of either of the two sets of bus-bars shown in Figure 1. As stated above, this loop of the sine wave will be affected by the firing of the tubes connected to the different lines of the bus. For example, if we number the terminals in the two Y's of the generator, 1, 2, 3 and 4, 5, 6, and if the loop of e.m.f. shown is across the terminals 1-2, then, as shown in Figure 3, it will be affected by the firing of the tubes across the other terminals as well as by those across terminals 1-2. Hence we have three jogs in each loop of the sine wave.

It will be noted that the jogs in the e.m.f. curve across terminals 1-2 are caused by e.m.f. curves 60 degrees in phase ahead of and behind the given curve respectively, instead of by e.m.f. curves 120 degrees in phase from the given curve as might be expected, since the phases are 120 degrees apart. The reason for this is that when we commutate from phase 2 to phase 3, for example, we have available, for commutating purposes, the difference in voltage between curves 1-2 and 1-3 and not curves 1-2 and 3-1. In each case, the voltage curve will change to a sine wave half way between the two sine waves involved. These three changes or jogs are shown in the figure. In the first one, the e.m.f. drops to zero because the e.m.f. of the other wave is exactly equal and opposite to it, or 180 degrees from it in phase; in other words, this represents a shift of tubes from one to the other of the two phases across which we are measuring e.m.f. The second jog is the result of a wave 60 degrees ahead of the one shown, and the third jog, the result of one 60 degrees behind the curve shown. The curve of e.m.f. across the two terminals will, therefore, be a b c d e f g h k l m n o p r, as shown in Figure 3.

As in the case of Figure 2, the angle α is the retardation angle of the grid and the angle ψ is the overlap of two tubes. The angle ψ must be the same in all three cases since the circuit is in all respects symmetrical. The maximum voltage across any two phases is represented by $E_{3\ ph\ max}$ as shown.

Since we wish to obtain an equation in terms of some value that can be measured, we must develop the equation to use the R.M.S.

(root of mean square) value of the e.m.f. across any two terminals, since that is what we measure—in other words, we must use the square root of the mean square value of the e.m.f. To obtain this, we must find the area included under a curve represented by the squares of the ordinates of the curve shown. If we divide this area by π and extract the square root, we shall have the R.M.S. value of the e.m.f.

As in the case of Figure 2, it will be best to calculate the areas separately and then add them together.

The first area to be obtained is that enclosed between the sine wave whose ordinates are the squares of those given by the curve a b, and the base line. The area of this curve will be given by the integral

$$\int_{0^\circ}^{\alpha} E^2_{3ph\ max} \sin^2 \theta \, d\theta$$

The next area will be that enclosed by the squares of the ordinates of curve e f g and this will be given by the integral

$$\int_{\alpha + \psi}^{60^\circ + \alpha} E^2_{3ph\ max} \sin^2 \theta \, d\theta$$

The third area will be that enclosed by the squares of the ordinates of the curve h k and this will be given by the integral

$$\int_{60^\circ + \alpha}^{60^\circ + \alpha + \psi} E^2_{3ph\ max} \left[1/4 \left\{ \sin \theta + \sin (\theta + 60^\circ) \right\}^2\right] d\theta,$$

this being taken from a sine wave half way between the sine waves given by $E_{3\ ph\ max} \sin \theta$ and $E_{3\ ph\ max} \sin (\theta + 60^\circ)$. The fourth area will be that enclosed by the squares of the ordinates of the curve l m and this will be given by the integral

$$\int_{60^\circ + \alpha + \psi}^{120^\circ + \alpha} E^2_{3ph\ max} \sin^2 \theta \, d\theta$$

The fifth area will be that enclosed by the squares of the ordinates of the curve n o and this will be given by the integral

$$\int_{120^\circ + \alpha}^{120^\circ + \alpha + \psi} E^2_{3ph\ max} \left[1/4 \left\{ \sin\theta + \sin(\theta - 60^\circ) \right\}^2\right] d\theta$$

The curve n o is a sine wave half way between the sine waves given by $E_{3\ ph\ max} \sin\theta$ and $E_{3\ ph\ max} \sin(\theta - 60^\circ)$. The sixth area will be that enclosed by the squares of the ordinates of the curve p r and this will be given by the integral

$$\int_{120^\circ + \alpha + \psi}^{180^\circ} E^2_{3ph\ max} \sin^2\theta\, d\theta$$

We will omit the several pages of integration necessary to get the values of these integrals, but when they are obtained and added together, we shall find that the total area is

$$E^2_{3ph\ max} \left[3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4} \right]$$

If we divide this by π to get the average value, and extract the square root, we shall have

$$E_L = \frac{E_{3ph\ max} \sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}}{\sqrt{\pi}} \qquad (2)$$

E_L is used to designate this value and it is the e.m.f. that we should read on a voltmeter placed across the terminals.

In equation (1), we found that

$$E_{dc} = \frac{3\sqrt{3}\, E_{G\ ph\ max}}{4\pi} \left[\cos(\alpha + \psi) + \cos\alpha\right]$$

But $E_{3\ ph\ max} = \sqrt{3}\, E_{G\ ph\ max}$, since this is a 3-phase machine. Substituting this value in the equation previously deduced for E_{dc} we have—

$$E_{dc} = \frac{3\sqrt{3}}{4\pi} \frac{E_{3ph\ max}}{\sqrt{3}} [\cos(\alpha + \psi) + \cos\alpha]$$

If we substitute the value of $E_{3\ ph\ max}$ given by Equation (2), we shall have—

$$E_{dc} = \frac{\frac{3\sqrt{3}}{4\pi} \times \frac{1}{\sqrt{3}} \times [\cos(\alpha + \psi) + \cos\alpha] \times \sqrt{\pi} \times E_L}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}}$$

or

$$E_{dc} = \frac{3\ E_L}{4\sqrt{\pi}} \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}} \quad (3)$$

We now have the value of E_{dc} in terms of a voltage EL, which can be measured with a voltmeter. However, we still can not know the value of E_{dc} from this equation since the value of ψ is unknown. The value of α is, of course, known for any given condition, since it is set, or fixed, by the operator; that is, he will always know at what value he has set α, as well as the measured value of EL; therefore, we need only to know ψ to complete the equation.

As explained above, when we shift from one tube to another, the local circuit formed by the two tubes will be a short circuit. The difference in voltage between that on the tube coming in and that on the tube going out will be available for reducing the current in the outgoing tube to zero and for building up the current in the incoming tube to load current. The amount of time taken for this will depend on this difference of voltage, on the reactance of the circuit, and on the amount of current flowing in the tube. This current is a direct current and we will designate it as I_{dc}; it is the value of the current flowing in the field of the motor. We will designate the reactance of one phase of the generator by XG.

The counter e.m.f. generated by the incoming current will be given by the equation, $e = L\frac{di}{dt}$. This e.m.f. must be equal to one-

half that given by the sine wave from b to e in Figure 3, since that is the voltage that is being used to build up the incoming current. One-half of it is being used in each half of the circuit consisting of two tubes and two generator phases.

Therefore we have—

$$L \frac{di}{dt} = 1/2\ E_{3ph\ max} \sin \theta$$

and $dt = \frac{d\theta}{\omega}$ since $\omega = 2\pi f$, f being the frequency

$$\therefore \omega L \frac{di}{d\theta} = 1/2\ E_{3ph\ max} \sin \theta$$

$$\text{or } \omega L\ di = 1/2\ E_{3ph\ max} \sin \theta\ d\theta$$

Integrating, we have—

$$\omega L \int_0^{I_{dc}} di = 1/2\ E_{3ph\ max} \int_{\alpha}^{\alpha + \psi} \sin \theta\ d\theta$$

and $\omega L I_{dc} = -1/2\ E_{3ph\ max} [\cos(\alpha + \psi) - \cos \alpha]$

But $\omega L = X_G$ and $E_{3ph\ max} = \sqrt{2} \times \sqrt{3}\ E_{ph}$

Substituting, we have—

$$\cos(\alpha + \psi) = \cos \alpha - \frac{X_G\ I_{dc}}{1/2\ E_{3ph\ max}}$$

$$\text{and } \cos(\alpha + \psi) = \cos \alpha - \frac{\sqrt{2}\ X_G\ I_{dc}}{\sqrt{3}\ E_{ph}} \qquad (4)$$

The second member of the right-hand part of the equation will be known for each generator and α is also known, so Equations (3) and (4) are sufficient to obtain the value of E_{dc}. As will be seen later, a different form of Equation (4) will ordinarily be used for the "full power" condition and Equation (4) will be used at "reduced powers."

We shall next find the equation for the value of the alternating current in the generator. The alternating current in the generator is shown in Figure 4. Figure 4a shows the current as it would be if there were no overlap of current in the tubes. Figure 4b shows

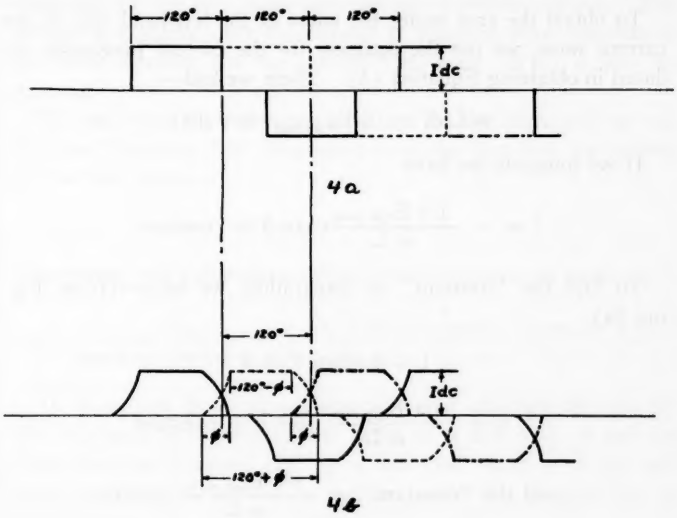

FIGURE 4.

the current as it actually is. To understand better Figure 4b, it should be placed under Figure 3. One of the current loops is thus shown in Figure 3a. This figure shows clearly the relations between current and voltage. The current in Figure 3a begins to flow in the given phase at the time that the voltage falls from the point b to the point c in Figure 3. Then when the current has reached its full value of I_{dc}, the voltage rises from the point d to the point e in Figure 3. The current is interrupted in this phase at the point m in Figure 3; it begins to decay and reaches zero at point o in Figure 3. The total period during which current flows in the given phase is the angle 120 degrees $+ \psi$ as shown in Figure 3a. To get the R.M.S. value of this current it will be necessary to find the areas under the curves formed by the squares of the ordinates of the three parts of the current wave shown in Figure 4b; this cur-

rent wave is composed of two curves and one straight line. The area under the straight line will be—

$$I_{dc}^2\ (120° - \psi) \text{ or } I_{dc}^2\ (\frac{2\pi}{3} - \psi)$$

To obtain the area under the curve at the left-hand side of the current wave, we use the equation for the current previously deduced in obtaining Equation (4). There we had—

$$\omega L\, di = \tfrac{1}{2} E_{3\ ph\ max} \sin\theta\, d\theta$$

If we integrate we have

$$i = -\frac{1/2\ E_{3ph\ max}}{\omega L} \cos\theta + \text{constant}$$

To find the "constant" of integration, we have—(from Figure 3a)

$$i = 0 \text{ when } \theta = \alpha$$

$$\therefore o = \frac{1/2\ E_{3ph\ max}}{\omega L} \cos\alpha + \text{constant}$$

$$\text{and the "constant"} = \frac{1/2\ E_{3ph\ max}}{\omega L} \cos\alpha$$

$$\text{and } i = \frac{1/2\ E_{3ph\ max}}{\omega L} [\cos\alpha - \cos\theta]$$

From Equation (4), we have

$$\frac{1/2\ E_{3ph\ max}}{\omega L} = \frac{I_{dc}}{\cos\alpha - \cos(\alpha + \psi)}$$

Substituting, we have

$$i = \frac{I_{dc}}{\cos\alpha - \cos(\alpha + \psi)} [\cos\alpha - \cos\theta]$$

For the right-hand curve of the current loop, we have—

$$L\frac{di}{dt} = -e = -1/2\ E_{3ph\ max} [\sin(\theta - 60°) - \sin\theta]$$

We obtain this equation from the fact that the e.m.f. available for reducing the current to zero is equal to the difference between the e.m.fs. given by the curves sin $(\theta - 60°)$ and sin θ in Figure 3 and one-half of the difference is used in each half of the circuit composed of the two tubes and the two generator phases. The voltage is negative since the current is decaying here, whereas in the other equation for the current (left-hand curve) it was increasing.

We can integrate this equation as we did the other and we can find the "constant" of integration from the fact that the current will be zero when—

$$\theta = 120° + \alpha + \psi$$

Substituting this value and simplifying the equation, we have—

$$i = \frac{I_{dc}}{\cos \alpha - \cos (\alpha + \psi)} \left[\frac{\sqrt{3} \sin \theta - \cos \theta - 2 \cos (\alpha + \psi)}{2} \right]$$

If we square these current equations and integrate the one for the left-hand curve between the limits of α and $\alpha + \psi$ and the right-hand one between $120° + \alpha$ and $120° + \alpha + \psi$ and add them together and add this to the area under the straight line we shall have—

$$\text{Total area} = \int_{\alpha}^{\alpha + \psi} \frac{I_{dc}^2}{[\cos \alpha - \cos (\alpha + \psi)]^2} [\cos \alpha - \cos \theta]^2 \, d\theta$$

$$+ \int_{120° + \alpha}^{120° + \alpha + \psi} \frac{I_{dc}^2}{[\cos \alpha - \cos (\alpha + \psi)]^2} \left[\frac{\sqrt{3} \sin \theta - \cos \theta - 2 \cos (\alpha + \psi)}{2} \right]^2 d\theta$$

$$+ I_{dc}^2 \left(\frac{2\pi}{3} - \psi \right)$$

Since we are dealing with only one Y of the generator, we must take only the areas of the current loops, either above or below the base line; the other loops will belong to the other Y. Therefore, to get the R.M.S. value of the current, we must divide the total area found above by 2π (there being only one current loop for 360 degrees) and then extract the square root; we shall then have—

$$I_{ac} = \frac{I_{dc}}{\sqrt{2\pi}} \sqrt{\psi\,[1 + \cos^2\alpha + \cos^2(\alpha+\psi)] - 1/2\,[\sin(2\alpha+2\psi) - \sin 2\alpha] - 2\sin\psi \over [\cos\alpha - \cos(\alpha+\psi)]^2 + \left[\frac{2\pi}{3} - \psi\right]} \tag{5}$$

This is the value of the actual current flowing in each Y of the generator, but since each slot of the generator will contain the winding of both Ys, we shall get the effect (so far as reactance is concerned) of having a single Y with the currents flowing in it as represented by the loops of Figure 4b, both above and below the base line. If we denote this slot current by I_{acs}, we can obtain its value by dividing the area found above by π (instead of 2π) and extracting the square root as before. We shall have—

$$I_{acs} = \frac{I_{dc}}{\sqrt{\pi}} \sqrt{\psi[1 + \cos^2\alpha + \cos^2(\alpha+\psi)] - 1/2\,[\sin(2\alpha+2\psi) - \sin 2\alpha] - 2\sin\psi \over [\cos\alpha - \cos(\alpha+\psi)]^2 + \left[\frac{2\pi}{3} - \psi\right]} \tag{6}$$

The relation between slot current and current in one of the Ys will be—

$$I_{ac} = \frac{I_{acs}}{\sqrt{2}} \tag{7}$$

The equation for slot current is useful because it is the slot current which determines the reactance of the generator. The value of I_{acs} can be substituted in Equation (4) and gives another method of solving for the angle ψ when the generator reactance is known.

In solving Equation (6), we shall find that, at full load, when α is usually zero and ψ is about 30 degrees, we shall have—

$$I_{acs} = .967\ I_{dc}\ \sqrt{2/3} \qquad (8)$$

If we substitute this value in Equation 4, we shall have (at full power)—

$$\cos(\alpha + \psi) = \cos\alpha - \frac{\sqrt{2}}{\sqrt{3}} \frac{X_G\ I_{acs}\ \sqrt{3}}{E_{ph}\ \sqrt{2} \times .967}$$

$$\text{or } \cos(\alpha + \psi) = \cos\alpha - \frac{X_G\ I_{acs}}{.967\ E_{ph}} \qquad (9)$$

This equation can be used instead of Equation (4) for solving for the angle ψ, when the generator reactance is known or assumed. Since this value can readily be measured, it will generally be more convenient to use Equation (9) than Equation (4) for the full power condition. At fractional loads, Equation (4) will be more convenient since the second part of the right-hand side of the equation will vary directly as the value of I_{dc}. For example, for the ordinary generator we can assume that the reactance will be about 12.95 per cent—that is the value of $\frac{X_G\ I_{acs}}{E_{ph}}$ is .1295. Therefore the value of $\frac{X_G\ I_{acs}}{.967\ E_{ph}}$ will be .134.

If we substitute this value in Equation (9) and also remember that α has been assumed to be zero, we have

$$\cos\psi = 1 - .134 = .866$$

and

$$\psi = 30 \text{ degrees} \qquad (10)$$

If we substitute these values in Equation (3), we shall have

$$E_{dc} = .645\ E_L \ (\textit{Full Power}) \qquad (11)$$

We shall next develop the equations for the series circuit thyratron motor. This is shown in Figure 5.

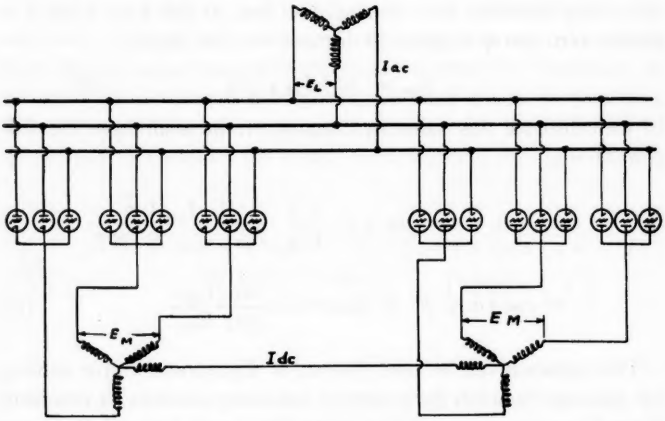

FIGURE 5.

It will be noted that, in most instances, the values here are exactly twice those for the parallel type of motor. There are some exceptions, however. We shall consider each equation separately.

First, for E_{dc} the value will be twice that of Figure 1 since the tube banks for the two halves of the motor are in series. Therefore we have

$$E_{dc} = \frac{3}{2}\frac{E_L}{\sqrt{\pi}} \times \frac{\cos(\alpha + \psi) + \cos\alpha}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}} \tag{12}$$

The current equation is identical with that used for finding Equation (4) and we can therefore write down the equation for finding the angle ψ as—

$$\cos(\alpha + \psi) = \cos\alpha - \frac{\sqrt{2}}{\sqrt{3}} \frac{X_G I_{dc}}{E_{ph}} \tag{13}$$

This is identical with Equation (4).

In finding the equation for alternating current in the generator, the situation is somewhat simpler in Figure 5 than it is in Figure 1. In Figure 5, the current in the slot is the same as that in the Y and

therefore Figure 4b represents the current in the Y as well as in the slot. The area of the squares of the current loops will therefore have to be divided by π (instead of 2π) before extracting the square root. We shall have

$$I_{ac} = \frac{I_{dc}}{\sqrt{\pi}} \sqrt{\frac{\psi\,[1 + \cos^2 \alpha + \cos^2 (\alpha + \psi)] - 1/2\,[\sin (2\alpha + 2\psi) - \sin 2\alpha] - 2 \sin \psi + \left[\frac{2\pi}{3} - \psi\right]}{[\cos \alpha - \cos (\alpha + \psi)]^2}} \tag{14}$$

Equation (14) is identical with Equation (6). Therefore the value of I_{ac} at full power will be the same as Equation (8) and we shall have

$$I_{ac} = .967\, I_{dc} \sqrt{2/3} \tag{15}$$

$$\text{also } \cos (\alpha + \psi) = \cos \alpha - \frac{\sqrt{2}}{\sqrt{3}} \frac{X_G\ I_{ac} \sqrt{3}}{\sqrt{2}\, E_{ph} \times .967}$$

$$\text{or } \cos (\alpha + \psi) = \cos \alpha - \frac{X_G\, I_{ac}}{.967\ E_{ph}}$$

If we use the same value of reactance as previously, we shall have

$$\cos \psi = 1 - .134$$

$$\text{or} \qquad \psi = 30 \text{ degrees} \tag{16}$$

If we substitute these values in Equation (12), we shall have—

$$E_{dc} = 1.29\, E_L \tag{17}$$

The above equations for series and parallel circuits are for the rectifier side of the tubes and give the values of E_{dc} in terms of the line voltage. The equations for the inverter, or motor, side of the tubes, can be found in a similar manner; however, they can be derived directly and more easily from the rectifier formulae.

If we number the motor terminals in the same manner as we did the generator, then Figure 6 will represent the curve of e.m.f. across two motor terminals such as 1-2 and shows how this curve

is modified by the firing of the tubes across the other terminals. For clarity, both halves of the curve are shown. The jogs in the e.m.f. curve are due to the same reasons as those given for the curve in Figure 3 and the explanation given for that figure also holds good here. It will be noted that the jogs in the e.m.f. curve

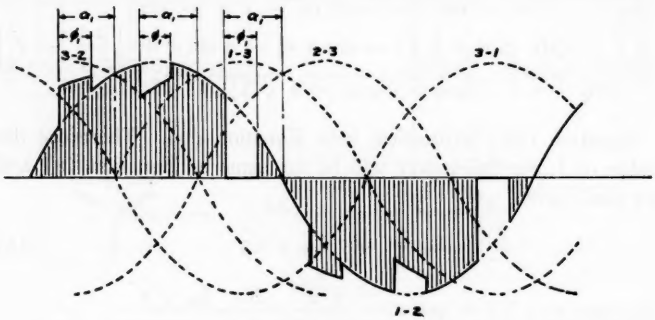

FIGURE 6.

are caused by e.m.f. curves 60 degrees in phase ahead of and behind the given curve, respectively for the reason that was given for Figure 3. It will be noted that the areas to be added to or deducted from the curve of e.m.f. are identical with those for the rectifier shown in Figure 3, assuming the angles to be equal in the two cases; the only difference between the two figures is that the areas in Figure 3 when read from left to right are identical with those in Figure 6 when the latter are read from right to left. Since we are dealing with an inverter we shall have an angle of advance instead of an angle of retardation and, therefore, we measure the angles to the left instead of to the right. In order to keep the two conditions separate, we shall denote the angle of advance by α_1 and the angle of overlap by ψ_1. We shall denote the terminal voltage by EM instead of EL, since it is the voltage measured across the motor terminals. The equivalent direct current voltage we shall designate as $E_{dc}M$ since it represents the voltage drop across the motor. This voltage will differ from E_{dc} only by the amount of the drop across a tube, which is small and which we shall designate as ET. From Figures 3 and 6, we note that

$$\alpha_1 = \alpha + \psi$$

If we substitute this value in Equation (3), and also substitute E_{dcM}, E_M and ψ_1 for E_{dc}, E_L and ψ, respectively, we shall have

$$E_{dcM} = \frac{3}{4\sqrt{\pi}} \sqrt{\frac{[\cos(\alpha_1 - \psi_1) + \cos\alpha_1]}{3/8 \sin 2\alpha_1 - 3/8 \sin(2\alpha_1 - 2\psi_1) - \frac{3\psi_1}{4} + \pi/2}} \quad (18)$$

Also, by substituting in Equation (4), we get

$$\cos(\alpha_1 - \psi_1) = \cos\alpha_1 + \frac{\sqrt{2}}{\sqrt{3}} \frac{X_M I_{d_c}}{E_{Mph}} \quad (19)$$

also $$E_{dc} = E_{dcM} + E_T \quad (20)$$

We can assume that motor and generator characteristics are similar. Therefore, at full power, the reactance factor will be equal to .134 and we shall have for full power—

$$\cos(\alpha_1 - \psi_1) = \cos\alpha_1 + .134 \quad (21)$$

As explained in the second paper on motors, the value of α_1 is fixed by the setting of the motor brushes and is usually constant. It will vary from 30 degrees to 40 degrees. If we assume a value of 30 degrees, then Equation (20) becomes—

$$\cos(30 \text{ degrees} - \psi_1) = .866 + .134 = 1$$

and $$30 \text{ degrees} - \psi_1 = 0$$

or $$\psi_1 = 30 \text{ degrees} \quad (22)$$

If we substitute these values in Equation (18) we shall have

$$E_{dcM} = .645\ E_M \quad (23)$$

We can obtain the equation for motor current by substituting in Equations (5) and (6) the values given above—

$$\alpha_1 = \alpha + \psi$$

We shall have—

$$I_{ac\,M} = \frac{I_{dc}}{\sqrt{2}\pi}\sqrt{\frac{\psi_1\,(1+\cos^2\alpha_1+\cos^2(\alpha_1-\psi_1)-1/2\,[\sin 2\alpha_1-\sin(2\alpha_1-2\psi_1)]-2\sin\psi_1+\left[\frac{2\pi}{3}-\psi\right]}{[\cos(\alpha_1-\psi_1)-\cos\alpha_1]^2}}$$

and

$$I_{acs\,M} = \frac{I_{dc}}{\sqrt{\pi}}\sqrt{\frac{\psi_1\,[(1+\cos^2\alpha_1+\cos^2(\alpha_1-\psi_1)]-1/2\,[\sin 2\alpha_1-\sin(2\alpha_1-2\psi_1)]-2\sin\psi_1+\left[\frac{2\pi}{3}-\psi\right]}{[\cos(\alpha_1-\psi_1)-\cos\alpha_1]^2}}$$

The inverter equations for Figure 5 can be obtained in the same way that we have obtained those above. We shall have—

$$E_{dc\,M} = \frac{3\,E_M}{2\sqrt{\pi}}\,\frac{[\cos(\alpha_1-\psi_1)+\cos\alpha_1]}{\sqrt{3/8\sin 2\alpha_1-3/8\sin(2\alpha_1-2\psi_1)-\frac{3\psi_1}{4}+\pi/2}} \qquad (24)$$

$$\cos(\alpha_1-\psi_1) = \cos\alpha_1+\frac{\sqrt{2}\,X_M\,I_{dc}}{\sqrt{3}\,E_{M\,ph}} \qquad (25)$$

If $\alpha_1 = 30$ degrees, then we shall have

$$\psi_1 = 30 \text{ degrees} \qquad (26)$$

If we substitute these values in Equation (24), we shall have

$$E_{dc}M = 1.29\,E_M \qquad (27)$$

In the above equations for parallel and series circuits, we have assumed the value of α_1 to be 30 degrees; also a reactance value, at full load, of 12.95 per cent, has been assumed in all cases; then the minimum possible value of α_1 is 30 degrees, since any smaller value of α_1 will give a value of cos $(\alpha_1 - \psi_1)$ greater than unity which is, of course, impossible. Therefore, the minimum value of α_1 is fixed by the reactance of the motor. When the value of α_1 becomes

approximately 90 degrees, the value of $E_{dc}M$ will become zero. α_1 will sometimes be slightly greater than 90 degrees owing to the overlap of the currents when shifting tubes.

The value of ψ_1 can never be greater than that of α_1 as shown by the equation $\cos(\alpha_1 - \psi_1) = \cos \alpha_1 + \frac{X_M I_{ac}}{E_{M\,ph}}$; if conditions were such as to cause it to be greater, the tube would not function. The minimum value is zero.

The conditions that govern the values of ψ are as follows. The minimum value is zero. ψ can not be greater than 60 degrees with a series three-phase arrangement, since any larger angle would give more than 180 degrees for the three tubes for any one loop of e.m.f. This angle will actually be less than 40 degrees as a rule. The value, in all cases, will depend on the reactance of the generator and the amount of load current, as shown by the formulae. For a parallel three-phase arrangement, the angle ψ really has no maximum limit, but actually it is usually less than 40 degrees.

The minimum value of α is 0 degrees. When α becomes approximately 90 degrees, E_{dc} will become 0 degrees. α can be slightly greater than 90 degrees owing to the effect of the overlapping of the currents, as shown in Figure 4b.

It is desirable now to collect the equations derived for the two circuits.

PARALLEL CONNECTION.

General (Rectifier).

$$E_{dc} = \frac{3\,E_L}{4\sqrt{\pi}} \frac{[\cos(\alpha + \psi) + \cos \alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}}$$

$$\cos(\alpha + \psi) = \cos \alpha - \frac{\sqrt{2}\, X_G\, I_{dc}}{\sqrt{3}\, E_{ph}}$$

Full Power (Rectifier).

$$\alpha = 0$$

$$\frac{X_G\, I_{ac}}{E_{ph}} = .1295$$

We shall have—

$$I_{ac\,M} = \frac{I_{dc}}{\sqrt{2}\pi} \sqrt{\frac{\psi_1 (1 + \cos^2 \alpha_1 + \cos^2 (\alpha_1 - \psi_1) - 1/2 [\sin 2\alpha_1 - \sin (2\alpha_1 - 2\psi_1)] - 2 \sin \psi_1 + \left[\frac{2\pi}{3} - \psi\right]}{[\cos (\alpha_1 - \psi_1) - \cos \alpha_1]^2}}$$

and

$$I_{acs\,M} = \frac{I_{dc}}{\sqrt{\pi}} \sqrt{\frac{\psi_1 [(1 + \cos^2 \alpha_1 + \cos^2 (\alpha_1 - \psi_1)] - 1/2 [\sin 2\alpha_1 - \sin (2\alpha_1 - 2\psi_1)] - 2 \sin \psi_1 + \left[\frac{2\pi}{3} - \psi\right]}{[\cos (\alpha_1 - \psi_1) - \cos \alpha_1]^2}}$$

The inverter equations for Figure 5 can be obtained in the same way that we have obtained those above. We shall have—

$$E_{dc\,M} = \frac{3\,E_M}{2\sqrt{\pi}} \frac{[\cos (\alpha_1 - \psi_1) + \cos \alpha_1]}{\sqrt{3/8 \sin 2\alpha_1 - 3/8 \sin (2\alpha_1 - 2\psi_1) - \frac{3\psi_1}{4} + \pi/2}} \qquad (24)$$

$$\cos (\alpha_1 - \psi_1) = \cos \alpha_1 + \frac{\sqrt{2}\, X_M\, I_{dc}}{\sqrt{3}\, E_{M\,ph}} \qquad (25)$$

If $\alpha_1 = 30$ degrees, then we shall have

$$\psi_1 = 30 \text{ degrees} \qquad (26)$$

If we substitute these values in Equation (24), we shall have

$$E_{dc}M = 1.29\ E_M \qquad (27)$$

In the above equations for parallel and series circuits, we have assumed the value of α_1 to be 30 degrees; also a reactance value, at full load, of 12.95 per cent, has been assumed in all cases; then the minimum possible value of α_1 is 30 degrees, since any smaller value of α_1 will give a value of cos $(\alpha_1 - \psi_1)$ greater than unity which is, of course, impossible. Therefore, the minimum value of α_1 is fixed by the reactance of the motor. When the value of α_1 becomes

approximately 90 degrees, the value of $E_{dc}M$ will become zero. α_1 will sometimes be slightly greater than 90 degrees owing to the overlap of the currents when shifting tubes.

The value of ψ_1 can never be greater than that of α_1 as shown by the equation $\cos(\alpha_1 - \psi_1) = \cos \alpha_1 + \frac{X_M I_{ac}}{E_{M\,ph}}$; if conditions were such as to cause it to be greater, the tube would not function. The minimum value is zero.

The conditions that govern the values of ψ are as follows. The minimum value is zero. ψ can not be greater than 60 degrees with a series three-phase arrangement, since any larger angle would give more than 180 degrees for the three tubes for any one loop of e.m.f. This angle will actually be less than 40 degrees as a rule. The value, in all cases, will depend on the reactance of the generator and the amount of load current, as shown by the formulae. For a parallel three-phase arrangement, the angle ψ really has no maximum limit, but actually it is usually less than 40 degrees.

The minimum value of α is 0 degrees. When α becomes approximately 90 degrees, E_{dc} will become 0 degrees. α can be slightly greater than 90 degrees owing to the effect of the overlapping of the currents, as shown in Figure 4b.

It is desirable now to collect the equations derived for the two circuits.

PARALLEL CONNECTION.

General (Rectifier).

$$E_{dc} = \frac{3\,E_L}{4\sqrt{\pi}} \frac{[\cos(\alpha + \psi) + \cos \alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}}$$

$$\cos(\alpha + \psi) = \cos \alpha - \frac{\sqrt{2}\, X_G\, I_{dc}}{\sqrt{3}\, E_{ph}}$$

Full Power (Rectifier).

$$\alpha = 0$$

$$\frac{X_G\, I_{ac}}{E_{ph}} = .1295$$

$E_{dc} = .645\ E_L$

$\text{Cos}\ \psi = .866$

$\psi = 30$ degrees

$I_{ac} = .967\ I_{dc}\ \sqrt{1/3} = .557\ I_{dc}$

$I_{acs} = .967\ I_{dc}\ \sqrt{2/3} = .79\ I_{dc}$

General (Inverter).

$$E_{dc\,M} = \frac{3\ E_M}{4\sqrt{\pi}} \frac{[\cos(\alpha_1 - \psi_1) + \cos\alpha_1]}{\sqrt{3/8 \sin 2\alpha_1 - 3/8 \sin(2\alpha_1 - 2\psi_1) + \pi/2 - \frac{3\psi_1}{4}}}$$

$$\cos(\alpha_1 - \psi_1) = \cos\alpha_1 + \frac{\sqrt{2}\ X_M\ I_{dc}}{\sqrt{3}\ E_{M\,ph}}$$

$$E_{dc} = E_{dc\,M} + E_T$$

Full Power (Inverter).

$\alpha_1 = 30$ degrees

$\frac{X_M\ I_{acs}}{E_{M\,ph}} = .1295$

$E_{dc\,M} = .645\ E_M$

$\text{Cos}\ \psi_1 = .866$

$\psi_1 = 30$ degrees

SERIES CONNECTION

General (Rectifier).

$$E_{dc} = \frac{3\ E_L}{2\sqrt{\pi}} \frac{[\cos(\alpha + \phi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\phi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\phi}{4}}}$$

$$\cos(\alpha + \psi) = \cos\alpha - \frac{\sqrt{2}\ X_G\ I_{dc}}{\sqrt{3}\ E_{ph}}$$

Full Power (Rectifier).

$\alpha = 0$

$\frac{X_G\ I_{ac}}{E_{ph}} = .1295$

$E_{dc} = 1.29\ E_L$

$\cos \psi = .866$

$\psi = 30$ degrees

$I_{ac} = .967\ I_{dc}\ \sqrt{2/3} = .79\ I_{dc}$

General (Inverter).

$$E_{dc\,M} = \frac{3\ E_M}{2\sqrt{\pi}} \frac{[\cos(\alpha_1 - \psi_1) + \cos \alpha_1]}{\sqrt{3/8 \sin 2\alpha_1 - 3/8 \sin(2\alpha_1 - 2\psi_1) + \pi/2 - \frac{3\psi_1}{4}}}$$

$$\cos(\alpha_1 - \psi_1) = \cos \alpha_1 + \frac{\sqrt{2}}{\sqrt{3}} \frac{X_M\ I_{dc}}{E_{M\,ph}}$$

$E_{dc} = E_{dc\,M} + 2\ E_T$

Full Power (Inverter).

$\alpha_1 = 30$ degrees

$\frac{X_M\ I_{ac}}{E_{M\,ph}} = .1295$

$E_{dc\,M} = 1.29\ E_M$

$\cos \psi_1 = .866$

$\psi_1 = 30$ degrees

The above types of circuits are generally called 6-tube circuits, being 3-phase circuits with two tubes for each phase so that both positive and negative loops of the e.m.f. wave are utilized completely. However, there are circuits which make use of only one loop of the e.m.f. wave. Examples of such circuits are shown in Figure 7, one being a battery charging circuit and the other being one-half of the arrangement shown in Figure 1.

The e.m.f. curve for such a circuit is shown in Figure 8. The interval between the firing of the tubes is 120 degrees; therefore,

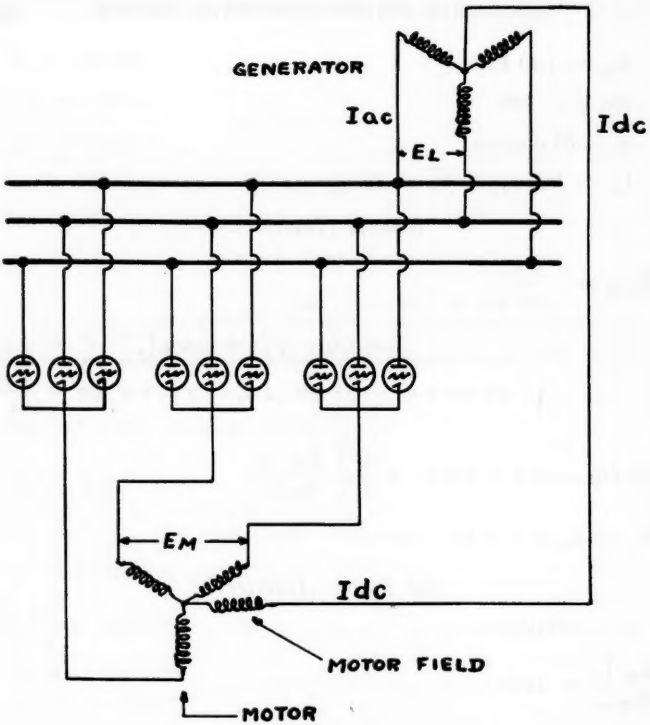

FIGURE 7-A.

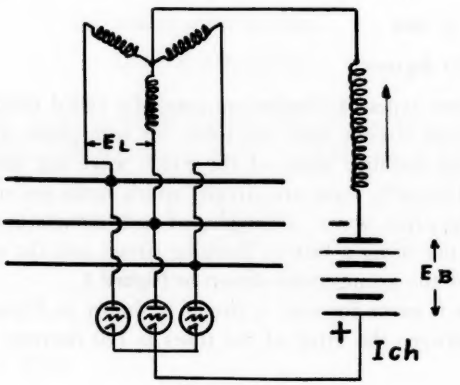

FIGURE 7-B.

FIGURE 7.

two of the tubes will fire in one loop of the wave and only one in the other. It will be noted that the two curves are different and there will, therefore, be a difference in the values of the positive and negative e.m.fs.

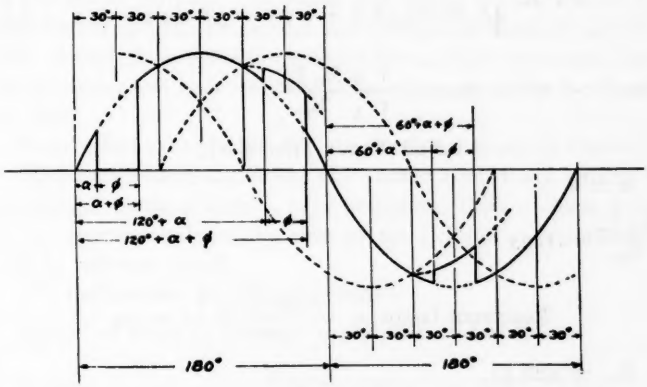

FIGURE 8.

The average value of the squares of the ordinates of these curves—in other words, the R.M.S. value of the e.m.f., can be found in the same manner as was used for the 6-tube circuits previously deduced, except that, in this case, we shall have to find the areas of two loops since the positive and negative loops are not the same. We do not need to go through all this integration, however, for, if we move the small subtractive area of the lower curve up to the upper curve and the corresponding portion of the upper curve down to the lower curve, the upper curve will be exactly the same as that of the 6-tube circuit for which we have already calculated the area, and the lower curve will be a complete sine wave, without irregularities; if we add the two areas together and divide by 2π (since we have two areas to average), we can deduce the desired equations very readily. Since the details are similar to the former procedure we shall spare the reader the boring labor of following them and the equations will be written directly.

3-TUBE CIRCUITS.

General (Rectifier).

$$E_{dc} = \frac{3\,E_L}{2\sqrt{2\pi}} \frac{[\cos(\alpha+\psi)+\cos\alpha]}{\sqrt{3/8\sin(2\alpha+2\psi) - 3/8\sin 2\alpha + \pi - \frac{3\psi}{4}}}$$

$$\cos(\alpha+\psi) = \cos\alpha - \frac{\sqrt{2}}{\sqrt{3}}\frac{X_G\,I_{dc}}{E_{ph}}$$

Full Power (Rectifier).

$\alpha = 0$

$$\frac{X_G\,I_{ac}}{E_{ph}} = .1295$$

$$\text{Reactance factor} = \frac{.1295 \times \sqrt{2}}{.961} = .19$$

$E_{dc} = .623\ E_L$

$\cos\psi = .81$

$\psi = 36$ degrees

$I_{ac} = .961\ I_{dc}\ \sqrt{1/3} = .553\ I_{dc}$

General (Inverter).

$$E_{dc\,M} = \frac{3\,E_M}{2\sqrt{2\pi}} \frac{[\cos(\alpha_1-\psi_1)+\cos\alpha_1]}{\sqrt{3/8\sin 2\alpha_1 - 3/8\sin(2\alpha_1-2\psi_1) + \pi - \frac{3\psi_1}{4}}}$$

$$\cos(\alpha_1-\psi_1) = \cos\alpha_1 + \frac{\sqrt{2}\,X_M\,I_{dc}}{\sqrt{3}\,E_{M\,ph}}$$

$$E_{dc} = E_{dc\,M} + E_T$$

Full Power (Inverter).

$\alpha_1 = 36$ degrees

$$\frac{X_M\,I_{ac}}{E_{ph}} = .1295$$

Reactance factor $= .19$

$E_{dc}M = .623\ E_M$

$\cos \psi_1 = .81$

$\psi_1 = 36$ degrees

The above formulae for 6-tube and 3-tube circuits should be sufficient to calculate the current and voltage conditions of any circuit. However, in order to make the use of them more clear, certain problems will be worked out to show how the formulae should be used.

For the first example, we will take the circuit shown in Figure 1.

Suppose we have, say, a 670 H.P. motor (500 Kw.), driving a centrifugal pump, a blower, or a propeller. Each of these loads varies approximately as the cube of the R.P.M. so one solution will be sufficient for all.

We shall assume the following data—

B.H.P. of motor at full load = 670 H.P. = 500 Kw.

Reactance of the motor $\frac{X_M\ I_{ac}}{E_{ph\ M}} = 12.95$ per cent $= .1295$

Generator or line voltage = 1000 volts

Motor eff. = 95 per cent = .95 (at full load)

Drop in volts through 1 tube = 10 volts.

Also that there is no retardation of the tubes when the motor is running at full load (500 Kw.)—that is, $\alpha = 0$, at that load.

The output of the motor $= E_{dc}M \times 2I_{dc} \times \text{Eff}_{motor} =$

$$E_{dc}M \times 2I_{dc} \times .95 = 500 \times 1000$$

But $$E_{dc} = .645E_L = .645 \times 1000 = 645 \text{ volts.} \quad \sharp$$

And $$E_{dc}M = E_{dc} - E_T = 645 - 10 = 635 \text{ volts.} \quad \sharp$$

$\therefore$ $$500 \times 1000 = 635 \times 2I_{dc} \times .95$$

and $$I_{dc} = \frac{500 \times 1000}{635 \times .95 \times 2} = \frac{828}{2} = 414 \quad \sharp$$

also $$E_M = \frac{635}{.645} = 985 \text{ volts.} \quad \sharp$$

Since there are 18 tubes, if we let $I_{T_{av}}$ and $I_{T_{max}}$ represent the average and maximum values of current per tube, we shall have

$$I_{T_{av}} = 828/18 = 46.0 \text{ amps.} \quad \sharp$$

and, since one-half of the total current flows through each bank of tubes, we shall have

$$I_{T_{max}} = 828/2 = 414 \text{ amps.} \quad \sharp$$

Also $$\cos(\alpha + \psi) = \cos\alpha - .134$$

Or $$\cos\psi = .866$$

And $$\psi = 30 \text{ degrees.} \quad \sharp$$

Suppose now that we desire to obtain the conditions of current, voltage, etc., for, say half-power. Assume that, at half-power,

$$\text{Motor eff.} = .93.$$

It will be necessary to make use of the laws governing load variation to obtain the complete solution. We have assumed a type of load which varies as the cube of the R.P.M. (which we shall represent by R). Also let T represent the torque and use the subscript "F.P." for full power conditions. Then we shall have—

$$\frac{R}{R_{FP}} = \sqrt[3]{\frac{SHP}{SHP_{FP}}} = \sqrt[3]{1/2}$$

and $$\frac{T}{T_{FP}} = \frac{R^2}{R_{FP}^2}$$

Also $$T = \text{constant} \times \text{field flux} \times I_{dc}$$

But the field flux $=$ constant $\times I_{dc}$, since the field flux varies as the field current, assuming that the field is not saturated at the full power condition

$$\therefore T = \text{constant} \times I_{dc}^2$$

and $$\frac{T}{T_{FP}} = \frac{I_{dc}^2}{I_{dc}^2{}_{FP}}$$

and $$\frac{I_{dc}^2}{I_{dc}^2{}_{FP}} = \frac{R^2}{R_{FP}^2}$$

$$\text{or } \frac{I_{dc}}{I_{dcFP}} = \frac{R}{R_{FP}} = \sqrt[3]{1/2}$$

$$\therefore I_{dc} = \frac{828}{2} \times \sqrt[3]{1/2} = 414 \times .794 = 328 \text{ amps. } \#$$

For the motor we have, under the above assumption of non-saturated field at full power, the condition that, with a constant value of α_1, the value of ψ_1 will also be constant regardless of load or speed. This will be seen from the following considerations. We have—

$$\text{Cos } (\alpha_1 - \psi_1) = \cos \alpha_1 + \frac{\sqrt{2}}{\sqrt{3}} \frac{X_M \; I_{dc}}{E_{M\,ph}}$$

But we have seen that

I_{dc} varies as the cube root of the horsepower (directly as the R.P.M.); but X_M will vary also directly as the R.P.M. and, therefore, as the cube root of the horsepower; also the value of $E_{M\,ph}$ will vary directly as the R.P.M. and also directly as the current. Therefore, the value of the reactance factor will be a constant since the various factors cancel each other. Since α_1 is also a constant, ψ_1 must be a constant. This will mean also that $E_{dc\,M} = .645\ E_M$ for all loads and speeds, as well as that of full power.

Since the motor output $= E_{dc\,M} \times I_{dc} \times \text{Eff}_M = \text{constant} \times R^3$

and $$I_{dc} = \text{constant} \times R$$

then $$E_{dc\,M} \times \text{Eff}_M = \text{constant} \times R^2$$

$$\text{and } \frac{E_{dc\,M} \times \text{Eff}_M}{E_{dc\,MFP} \times \text{Eff}_{MFP}} = \frac{R^2}{R_{FP}^2} = \frac{1}{2^{2/3}}$$

$$\therefore \frac{E_{dc\,M} \times .93}{635 \times .95} = \frac{1}{1.588}$$

$$\text{and } E_{dc\,M} = \frac{635 \times .95}{1.588 \times .93} = 409 \text{ volts } \#$$

also $E_{dc\,M} =$

$$\frac{3}{4}\frac{E_M}{\sqrt{\pi}} \frac{[\cos(\alpha_1 - \psi_1) + \cos\alpha_1]}{\sqrt{3/8 \sin 2\alpha_1 - 3/8 \sin(2\alpha_1 - 2\psi_1) + \pi/2 - \frac{3\psi_1}{4}}}$$

Substituting known values, we have

$$409 = .645\ E_M$$

and

$$E_M = 635 \text{ volts.} \quad \sharp$$

$$\text{Motor output (Kw.)} = \frac{409 \times 2 \times 328 \times .93}{1000} = 250 \text{ Kw.} = 335 \text{ H.P.} \sharp$$

also $$E_{dc\,M} = E_{dc} - E_T = E_{dc} - 10 = 409 \text{ volts}$$

and

$$E_{dc} = 419 \text{ volts.} \quad \sharp \quad \therefore$$

also $E_{dc} =$

$$\frac{3}{4}\frac{E_L}{\sqrt{\pi}} \times \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}}$$

$\therefore 419 =$

$$\frac{3 \times 1000}{4 \times 1.772} \times \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}}$$

and

$$\frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}} = \frac{419 \times 4 \times 1.772}{3 \times 1000} = .99$$

Also

$$\cos(\alpha + \psi) = \cos\alpha - .106.$$

From these last two equations, we can obtain the values of ψ and α which will be

$\alpha = 49.9$ degrees ♯

$\psi = 7.6$ degrees ♯

We can now collect these calculations, and we have—

Full Power.

B.H.P. (motor)	= 670 H.P.
Kw. output (motor)	= 500 Kw.
E_L	= 1000 volts
Motor eff.	= .95
E_T	= 10 volts
E_{dc}	= 645 volts
$E_{dc}M$	= 635 volts
E_M	= 985 volts
I_{dc}	= 414 amps.
$I_{T\,av}$	= 46 amps.
$I_{T\,max}$	= 414 amps.
α	= 0 degrees
ψ	= 30 degrees
α_1	= 30 degrees
ψ_1	= 30 degrees
Reactance	= .1295

Half Power.

B.H.P. (motor)	= 335 H.P.
Kw. output (motor)	= 250 Kw.
E_L	= 1000 volts
Motor eff.	= .93
E_T	= 10 volts
E_{dc}	= 419 volts
$E_{dc}M$	= 409 volts
E_M	= 635 volts
I_{dc}	= 328 amps.

$I_{T\,av}$	= 36.3 amps.
$I_{T\,max}$	= 328 amps.
α	= 49.9 degrees
ψ	= 7.6 degrees
α_1	= 30 degrees
ψ_1	= 30 degrees

Let us take the circuit shown in Figure 5 for our next calculations. For purposes of comparison, let us use the same data that we assumed for the previous problems.

Again we have for full power—

B.H.P. (Motor)	= 670 H.P.
Kw. output (motor)	= 500 Kw.
Reactance	= .1295
E_L	= 1000 volts
Motor eff.	= .95
E_T	= 10 volts
α	= 0 degrees.

We have then—

$$E_{dc} = 1.29\ E_L = 1.29 \times 1000 = 1290 \text{ volts.} \quad \sharp$$

Also $E_{dc\,M} = E_{dc} - 2\ E_T = 1290 - 20 = 1270$ volts. $\sharp$

$$\text{also } 500 = \frac{E_{dc\,M} \times I_{dc} \times Eff_{motor}}{1000} = \frac{1270 \times I_{dc} \times .95}{1000}$$

$$\therefore \quad I_{dc} = 414 \text{ amps.} \quad \sharp$$

$$I_{T_{av}} = 414/9 = 46 \text{ amps.} \quad \sharp$$

$$I_{T_{max}} = 414 \text{ amps.} \quad \sharp$$

Also $E_{dc\,M} = 1.29\ E_M$

$$\therefore E_M = \frac{1270}{1.29} = 985 \text{ volts} \sharp$$

Also $\cos(\alpha + \psi) = \cos\alpha - .134$

And $$\alpha = 0$$

$$\therefore \quad \cos \psi = .866$$

And $$\psi = 30 \text{ degrees.} \quad \#$$

For the half-power condition, we have, as before—

$$I_{dc} = 414 \times \sqrt[3]{1/2} = 328 \text{ amps.} \#$$

$$\text{also } \frac{E_{dc\,M} \times Eff_M}{E_{dc\,MFP} \times Eff_{MFP}} = \frac{1}{2^{2/3}}$$

$$\text{or } \frac{E_{dc\,M} \times .93}{1270 \times .95} = \frac{1}{1.588}$$

$$\text{and } E_{dc\,M} = \frac{1270 \times .95}{.93 \times 1.588} = 818 \text{ volts} \#$$

also $E_{dc\,M} =$

$$\frac{3}{2}\frac{E_M}{\sqrt{\pi}} \frac{[\cos(\alpha_1 - \psi_1) + \cos \alpha_1]}{\sqrt{3/8 \sin 2\alpha_1 - 3/8 \sin(2\alpha_1 - 2\psi_1) + \pi/2 - \frac{3\psi_1}{4}}}$$

Or $$818 = 1.29\ E_M$$

And $$E_M = 635 \text{ volts.} \quad \#$$

$$\text{Also motor output (Kw.)} = E_{dc\,M} \times I_{dc} \times Eff_M$$

$$= 818 \times 328 \times .93$$

$$= 250 \text{ Kw.}$$

$$= 335 \text{ H.P.} \quad \#$$

Also $$E_{dc\,M} = E_{dc} - 2\ E_T = E_{dc} - 20 = 818$$

$$\therefore \quad E_{dc} = 838 \text{ volts.} \quad \#$$

$$\text{Also } E_{dc} = \frac{3E_{L_c}}{2\sqrt{\pi}} \frac{[\cos(\alpha + \psi) + \cos \alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3\psi}{4}}}$$

$$\text{or } 838 = \frac{3 \times 1000}{2\sqrt{\pi}} \times \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3}{4}\psi}}$$

$$\text{And } \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi/2 - \frac{3}{4}\psi}} = \frac{2\sqrt{\pi} \times 838}{3 \times 1000} = .985$$

Also $$\cos(\alpha + \psi) = \cos\alpha - .106$$

From the last two equations, we obtain

$$\alpha = 49.9 \text{ degrees.} \quad \sharp$$

And $$\psi = 7.6 \text{ degrees.} \quad \sharp$$

Collecting the results obtained, we have—

Full Power.

B.H.P. (motor)	= 670 H.P.
Kw. output (motor)	= 500 Kw.
E_L	= 1000 volts
Motor eff.	= .95
E_T	= 10 volts
E_{dc}	= 1290 volts
$E_{dc}M$	= 1270 volts
E_M	= 985 volts
I_{dc}	= 414 amps.
$I_{T\,av}$	= 46 amps.
$I_{T\,max}$	= 414 amps.
α	= 0 degrees
ψ	= 30 degrees

α_1	= 30 degrees
ψ_1	= 30 degrees
Reactance	= .1295

Half Power.

B.H.P. (motor)	= 335 H.P.
Kw. output (motor)	= 250 Kw.
Motor eff.	= .93
E_T	= 10 volts
E_{dc}	= 838 volts
$E_{dc}M$	= 818 volts
E_M	= 635 volts
I_{dc}	= 328 amps.
$I_{T\,av}$	= 36.3 amps.
$I_{T\,max}$	= 328 amps.
α	= 49.9 degrees
ψ	= 7.6 degrees
α_1	= 30 degrees
ψ_1	= 30 degrees

It is interesting to compare the values obtained in the two cases of thyratron motor, first with parallel circuits and second with series circuits, where the horsepower is the same in both cases. It will be noted that the value of I_{dc} is the same in both cases which means that the current per tube is also the same in both cases. The total current, of course, is twice as much for the parallel circuit as for the series circuit and also the value of $E_{dc}M$ is twice as much for the series circuit as for the parallel circuit. Apparently, therefore, for ordinary conditions, neither type of motor has an advantage over the other so far as the current required per tube for a given horsepower is concerned. However, the series type of motor will ordinarily be used because it does not require a neutral return to the generator and because the slot current conditions in the generator will be more nearly perfect than in the case of the parallel type of motor. However, for certain

types of load, as will be seen later, the parallel circuit is necessary because the neutral return is a necessary part of the circuit.

Suppose we now take the circuit shown in Figure 7b for our next calculations. Assume that it is desired to maintain a charging current of 300 amperes and that this requires 350 volts across the battery. We then have—

$I_{ch} = I_{dc} = 300$ amps.

$E_B = 350$ volts.

For battery charging, the grids will generally be kept full open ($\alpha = 0$ degrees) and the voltage across the battery will be regulated by adjusting the field strength of the generator. It can also be done by maintaining a constant voltage on the generator and adjusting the angle of retardation of the grids; if the latter practice is used, the value of α will be greatest at the beginning of the charge (greatest load on the generator) and will decrease as the battery becomes partially charged, so that α will be smallest at the least load of the generator; for certain circuits, the latter method will be necessary. For this problem, however, we shall assume that the generator voltage is varied and that α is zero; the second method will be covered by another set of calculations.

For simplicity of calculations, we will assume that the generator shown in the figure has the same value of 12.95 per cent reactance so that we can use the "full power" equations given for the "3-tube" rectifier. We will have—

$\alpha = 0$ degrees.

$$\frac{X_G I_{ac}}{E_{ph}} = .1295.$$

$I_{dc} = I_{ch} = 300$ amps.

$E_B = 350$ volts.

$E_T = 10$ volts.

Then $E_{dc} = E_B + E_T = 350 + 10 = 360$ volts. ♯

Also $E_{dc} = .623\ E_L$.

And $E_L = 578$ volts. ♯

Also $I_{T\,av} = 100$ amps. ♯

And $I_{T\,max} = 300$ amps. ♯

For any lower value of charging current, say a tapering charge, where $I_{dc} = 50$ amperes, suppose we require a voltage across the battery of 400 volts. We shall then have

$\alpha = 0$ degrees.

And $\sqrt{2} \times$ Reactance $= \sqrt{2} \times .134 \times 50/300 = .032$.

Also Cos $(\alpha + \psi) = \cos \alpha - .032$.

And $\cos \psi = 1 - .032 = .968$.

And $\psi = 14$ degrees. ♯

Also $E_{dc} = 400 + 10 = 410$ volts. ♯

$$\text{Also } E_{dc} = \frac{3\ E_L}{2\sqrt{2}\pi} \times \frac{[\cos(\alpha + \psi) + \cos \alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \pi - \frac{3\psi}{4}}}$$

And $E_L = 617$ volts. ♯

Collecting the calculated values, we have

Full Charging Rate.

α	= 0 degrees
ψ	= 36 degrees
Reactance	= .1295
I_{dc}	= 300 amps.
$I_{T\,av}$	= 100 amps.
$I_{T\,max}$	= 300 amps.
E_B	= 350 volts
E_T	= 10 volts
E_{dc}	= 360 volts
E_L	= 578 volts

Tapering Charge.

α	= 0 degrees
ψ	= 14 degrees
$I_{dc} = I_{ch}$	= 50 amps.
$I_{T\,av}$	= 16.7 amps.
$I_{T\,max}$	= 50 amps.
E_B	= 400 volts
E_{dc}	= 410 volts
E_L	= 617 volts

It is believed that the above examples are sufficient to show, in general, how the calculations for any type of circuit can be made. There are, however, two combination circuits which are sufficiently unique to warrant making sample calculations. These are, first, a generator, used both for general purposes, such as driving a motor, and also for battery charging at the same time. The second example will be that of a battery, driving a motor through a generator, as an intermediary.

The first circuit is shown in Figure 9. The various current relationships are shown in the Figure. It will be noted that—

$$2\,I_{dc} = I_{ch} + I_G$$

or

$$I_G = 2\,I_{dc} - I_{ch}$$

In other words, the current returning to the generator neutral is the difference between the current flowing through the motor and that required to charge the battery. The actual path from the generator, through the battery, and through the tubes to the bus-bars is just the opposite of that from the generator through the motor, hence the generator neutral current will be the difference between the two.

The minimum possible voltage that can be used for this sort of a circuit will be that which will give the requisite voltage for charging the battery, but if the generator voltage is kept above this minimum, then the voltage across the battery can be regulated by the grids of the tubes supplying the battery. The motor speed can

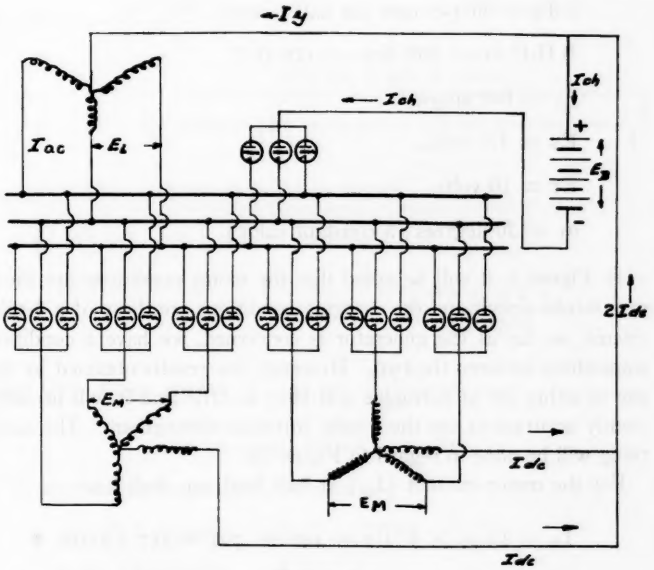

FIGURE 9.

likewise be regulated by proper adjustment of the grids of the tubes supplying the motor, so that the two parts of the circuits will be virtually independent of each other.

For our calculations, we will assume that the motor is operating at less than full power since the calculations for reduced power will include the steps required for full power. Let us assume that we want to have a voltage across the battery of 375 volts and that we require a charging current (I_{ch}) of 600 amperes; that the motor is operating at half-power; that the full power current of the generator is 900 amperes or $I_{dc} = 450$ amperes; that the line voltage is 700 volts and that the generator and motor reactances are each 12.95 per cent. We shall have—

$E_L = 700$ volts.

Reactance $= .1295$ at full power.

$$I_{dc\ F.P.} = \frac{900}{2} = 450 \text{ amps.}$$

$\text{Eff}_M = 90$ per cent (at half power).

B.H.P. $_{FP} = 350$ Kw. $= 470$ H.P.

$I_{ch} = 600$ amps.

$E_B = 375$ volts.

$E_T = 10$ volts.

$\alpha_1 = 30$ degrees (a constant value).

In Figure 9, it will be noted that the motor conditions are those of a 6-tube circuit and the charging conditions are those of a 3-tube circuit; so far as the generator is concerned, we have a condition somewhere between the two. However, the results obtained by the use of either set of formulae will vary so little that it will be sufficiently accurate to use the 6-tube formulae throughout. The same thing will be done in regard to Figure 10.

For the motor current (I_{dc}) at half load, we shall have—

$$I_{dc} = I_{dc\,FP} \times \sqrt[3]{1/2} = 450 \times .794 = 357.3 \text{ amps. \#}$$

And $I_G = (2 \times 357.3) - 600 = 714.6 - 600 = 114.6$ amps. #

$$\text{Also } \frac{E_{dc\,M} \times 2\, I_{dc} \times \text{Eff}_m}{1000} = 1/2 \times 350 = 175$$

$$\text{Or } \frac{E_{dc\,M} \times 714.6 \times .9}{1000} = 175$$

Or $E_{dc\,M} = 272$ volts #

Also $E_{dc} = E_{dc\,M} + E_T = 282$ volts #

Also $E_{dc\,M} = .645\, E_M$

Or $E_M = 422$ volts #

Also $E_{dc} =$

$$\frac{3\, E_L}{4\sqrt{\pi}} \times \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}}$$

$$\text{Or } 282 = \frac{3 \times 700}{4\sqrt{\pi}} \times \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}}$$

$$\text{Or } \frac{[\cos(\alpha + \psi) + \cos\alpha]}{\sqrt{3/8 \sin(2\alpha + 2\psi) - 3/8 \sin 2\alpha + \frac{\pi}{2} - \frac{3\psi}{4}}} = \frac{282 \times 4\sqrt{\pi}}{3 \times 700} = .952$$

$$\text{Also Gen. React. factor} = .134 \times \frac{714.6}{900} = .106$$

$$\therefore \quad \cos(\alpha + \psi) = \cos\alpha - .106.$$

From the above equations, we obtain—

$\alpha = 51.7$ degrees. ♯

$\psi = 7.4$ degrees. ♯

From the battery we have—

$E_B = 375$ volts.

E_{dc} (battery) $= 375 + 10 = 385$ volts. ♯

$$\text{Also } E_{dc\,bat} = \frac{3\,E_L}{4\sqrt{\pi}} \times \frac{[\cos(\alpha + \psi)_{bat} + \cos\alpha_{bat}]}{\sqrt{3/8 \sin(2\alpha + 2\psi)_{bat} - 3/8 \sin 2\alpha_{bat} + \pi/2 - \frac{3\psi_{bat}}{4}}}$$

$$\text{Or } \frac{[\cos(\alpha + \psi)_{bat} + \cos\alpha_{bat}]}{\sqrt{3/8 \sin(2\alpha + 2\psi)_{bat} - 3/8 \sin 2\alpha_{bat} + \pi/2 - \frac{3\psi_{bat}}{4}}} = \frac{385 \times 4\sqrt{\pi}}{3 \times 700} = 1.3$$

Also Reactance factor $= .134 \times 600/900 = .089.$

$\therefore \cos(\alpha + \psi)_{bat} = \cos \alpha_{bat} - .089$

From the above, we obtain—

$\alpha_{bat} = 28$ degrees. ♯

$\psi_{bat} = 9$ degrees. ♯

Collecting our calculations we have—

B.H.P. (motor)	= 235 H.P.
Kw. (output) (motor)	= 175 Kw.
E_L	= 700 volts
Motor eff.	= .9
E_T	= 10 volts
E_{dc}	= 282 volts
$E_{dc}M$	= 272 volts
$E_{dc\ bat}$	= 385 volts
E_M	= 422 volts
I_{dc}	= 357.3 amps.
I_{ch}	= 600 amps.
I_G	= 114.6 amps.
α (motor tubes)	= 51.7 degrees
ψ (motor tubes)	= 7.4 degrees
α_1	= 30 degrees
ψ_1	= 30 degrees
α (charging tubes)	= 28 degrees
ψ (charging tubes)	= 9 degrees

The second combined circuit is shown in Figure 10. Here we have a battery feeding current to the neutral of a generator, which, in turn, supplies a set of bus-bars; this bus supplies a motor and also a set of charging tubes for the battery. It is assumed that the generator is running when it is connected to the battery as there would be no torque for starting it. The current relations in the various parts of the circuit are shown. It will be noted that the generator neutral current is equal to the difference between motor

current and charging current as in the case of the circuit shown in Figure 9. In one important essential, however, the relationships of the various parts of this circuit differ from those of the circuit shown in Figure 9. The motor tubes will now be acting as inverters with respect to the generator, as well as with respect to the

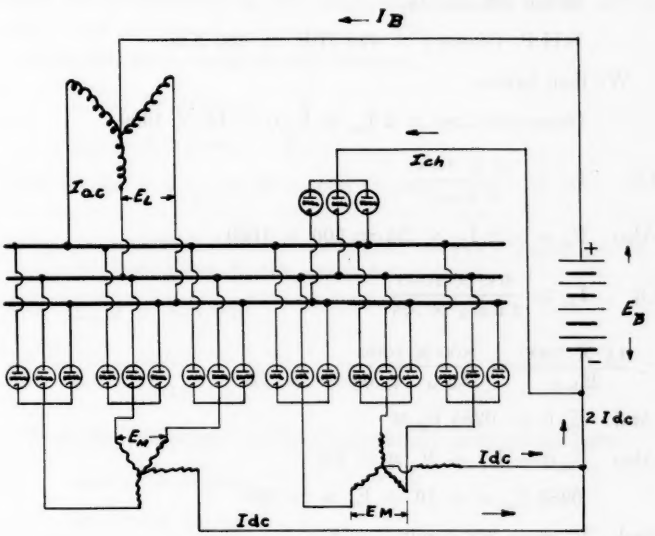

FIGURE 10.

motor. But the angle of advance and the angle of overlap will be different on the generator side from what they are for the motor side; also the angle of advance on the generator side will be variable while that of the motor will remain fixed, as before. The battery charging tubes will function in exactly the same manner as already described for the circuit shown in Figure 9.

For the first part of this problem, we will assume that the motor is running at full-power and that the charging tubes are entirely disconnected from the bus. In this case the entire current will flow through both generator and motor. For this condition, we shall assume the following data—

$E_L = 700$ volts

$E_B = 325$ volts

$E_T = 10$ volts

Generator Loss (Power absorbed) $= 15$ Kw.

Motor eff. $= .95$

B.H.P. (motor) $= 670$ H.P. $= 500$ Kw.

We then have—

$$\text{Generator Loss} = 2\, I_{dc} \times E_{dc}G = 15 \times 1000$$

$$\text{Or} \quad I_{dc} = \frac{15 \times 1000}{2\, E_{dc\,G}}$$

$$\text{Also} \quad E_{dc}M \times 2\, I_{dc} \times .95 = 500 \times 1000$$

$$\text{Or} \quad I_{dc} = \frac{500 \times 1000}{2\, E_{dc\,M} \times .95}$$

$$\therefore \frac{15 \times 1000}{E_{dc\,G}} = \frac{500 \times 1000}{E_{dc\,M} \times .95}$$

$$\text{And} \quad E_{dc}G = .0285\, E_{dc}M$$

$$\text{Also} \quad E_{dc}G + E_T + E_{dc}M = E_B$$

$$\therefore \quad .0285\, E_{dc}M + 10 + E_{dc}M = 325$$

$$\text{And} \quad E_{dc}M = 306.4 \text{ volts.} \quad \sharp$$

$$\text{And} \quad E_{dc}G = 8.6 \text{ volts.} \quad \sharp$$

$$\text{also } E_{dc\,G} = \frac{3\, E_L}{4\sqrt{\pi}} \times \frac{\cos(\alpha_1 - \psi_1)_G + \cos \alpha_{1G}}{\sqrt{3/8 \sin 2\,\alpha_{1\,G} - 3/8 \sin(2\,\alpha_1 - 2\,\psi_1)_G - \frac{3\,\psi_{1\,G}}{4} + \frac{\pi}{2}}}$$

$$\therefore 8.6 = \frac{3 \times 700}{4\sqrt{\pi}} \times \frac{\cos(\alpha_1 - \psi_1)_G + \cos \alpha_{1G}}{\sqrt{3/8 \sin 2\,\alpha_{1\,G} - 3/8 \sin(2\,\alpha_1 - 2\,\psi_1)_G - \frac{3\,\psi_{1G}}{4} + \frac{\pi}{2}}}$$

and

$$\frac{\cos\,(\alpha_1 - \psi_1)_G + \cos\,\alpha_{1\,G}}{\sqrt{3/8 \sin 2\,\alpha_{1\,G} - 3/8 \sin\,(2\,\alpha_1 - 2\,\psi_1)_G - \frac{3\,\psi_{1\,G}}{4} + \frac{\pi}{2}}} = .0291$$

Since we are operating at full power, the reactance factor will be .134 $\therefore \cos\,(\alpha_1 - \psi_1)_{\,G} = \cos\,\alpha_{1G} + .134$

From the above equations we obtain—

$\alpha_{1G} = 93$ degrees. ♯

$\psi_{1G} = 8$ degrees. ♯

Also $E_{dc}M \times 2\,I_{dc} \times .95 = 500 \times 1000$

$$\therefore I_{dc} = \frac{500 \times 1000}{2 \times 306.4 \times .95} = \frac{1715}{2} = 857.5 \text{ amps. } \#$$

Also $E_{dc}M = .645\,E_M$

$$\text{And } E_M = \frac{306.4}{.645} = 475 \text{ volts } \#$$

From the full power conditions previously deduced we have

$\alpha_{1\,M} = 30$ degrees. ♯

$\psi_{1M} = 30$ degrees. ♯

Collecting our calculations, we have, for the full-power condition—

B.H.P. (motor)	= 670 H.P.
Kw. (output) (motor)	= 500 Kw.
E_L	= 700 volts
Motor eff.	= .95
E_T	= 10 volts
$E_{dc}M$	= 306.4 volts
$E_{dc}G$	= 8.6 volts
E_B	= 325 volts
E_M	= 475 volts
Gen. Loss	= 15 Kw.

I_{dc} = 857.5 amps.
α_{1G} = 93 degrees
ψ_{1G} = 8 degrees
α_{1M} = 30 degrees
ψ_{1M} = 30 degrees

Now to run this circuit at reduced power, it will be necessary to cut in the charging tubes, so that any excess power, not required by the motors, will be returned to the battery. We shall now have the condition shown in Figure 10, with the charging tubes connected. The situation is rather complicated, but can be solved, since there must be a balance between generator voltage, generator current, charging current and motor current. The generator speed will automatically rise or fall with every change in tube setting; a balance will be struck when the voltage of the generator is just sufficient to give the necessary charging voltage to maintain the generator current at the requisite amount for its speed.

Let us make our calculations for the half-load conditions and assume that the motor efficiency is 93 per cent at this point.

Then $I_{dc} = 857.5 \times \sqrt[3]{1/2} = 857.5 \times .794 = 680$ amps. ♯

Also $E_{dcM} \times 2\, I_{dc} \times .93 = \frac{1}{2} \times 500 \times 1000$

$$\therefore \quad E_{dcM} = \frac{250 \times 1000}{1360 \times .93} = 197.5 \text{ volts.} \quad ♯$$

We shall now have to assume the battery characteristic. Let us assume that, if the battery voltage is 325 at the 1715 ampere rate, then the open circuit voltage will be 359.3 volts. Thus we have a rise of battery voltage of one volt for each 50 amperes reduction of battery current.

If the overall efficiency at the half-power condition is 97 per cent of what it is at the full power condition then we shall have—

$$\text{Battery output at half power} = \frac{1/2 \times 1715 \times 325}{.97}$$

$$\text{Also battery output at half power} = I_B \left[325 + \frac{1715 - I_B}{50} \right]$$

$$\therefore I_B \left[325 + \frac{1715 - I_B}{50}\right] = \frac{1/2 \times 1715 \times 325}{.97}$$

And $I_B = 840$ amps. ♯

Also $E_B = 325 + \frac{1715 - 840}{50} = 342.5$ volts ♯

And $I_{ch} = 1360 - 840 = 520$ amps. ♯

$E_{dc\ ch} = 342.5 + 10 = 352.5$ volts. ♯

Since α_{bat} for the charging tubes is 0 degrees, we have

$$\cos\ (\alpha_{bat} + \psi_{bat}) = \cos\ \alpha_{bat} - \frac{.134 \times 520}{1715}$$

$$\cos \psi_{bat} = 1 - .041 = .959$$

$$\psi_{bat} = 16.5 \text{ degrees}$$

And $E_{dc\ ch} = \frac{3\ E_L}{4\sqrt{\pi}} \times \frac{.959 + 1}{\sqrt{3/8 \times .544 + \pi/2 - 3/4 \times \frac{165}{180} \times \pi}}$

Or $352.5 = E_L \times .669$

And $E_L = 527$ volts. ♯

Battery output $= \frac{342.5 \times 860}{1000} = 294.5$ Kw. ♯

Motor input $= \frac{197.5 \times 1360}{1000} = 268.5$ Kw. ♯

Eff. of transmission $= \frac{268.5}{294.5} = 91.2$ per cent. ♯

Motor output $= .93 \times 268.5 = 250$ Kw. ♯

Overall eff. $= \frac{250}{294.5} = 84.8$ per cent. ♯

$$E_{dc}G = 342.5 - 10 - 197.5 = 135 \text{ volts.} \quad \#$$

$$\text{Loss in charging tubes} = \frac{10 \times 520}{1000} = 5.2 \text{ Kw.} \quad \#$$

$$\text{Loss in motor tubes} = \frac{10 \times 1360}{1000} = 13.6 \text{ Kw.} \quad \#$$

Total tube loss $= 5.2 + 13.6 = 18.8$ Kw.

Battery output — motor input $= 294.5 - 268.5 = 26$ Kw.

Gen. Loss $= 26 - 18.8$ Kw. $= 7.2$ Kw.

We can now find the values of α_1 and ψ_1 for both generator and motor.

For the generator we have—

$$135 = \frac{3 \times 527}{4\sqrt{\pi}} \times \frac{\cos(\alpha_{1G} - \psi_{1G}) + \cos \alpha_{1G}}{\sqrt{3/8 \sin 2\alpha_{1G} - 3/8 \sin(2\alpha_{1G} - 2\psi_{1G}) + \pi/2 - 3/4 \times \psi_{1G}}}$$

$$\text{and } \cos(\alpha_{1G} - \psi_{1G}) = \cos \alpha_{1G} + \frac{.134 \times 1360}{1715} = \cos \alpha_{1G} + .106$$

$\therefore \alpha_{1G} = 71.8$ degrees

And $\psi_{1G} = 6.5$ degrees #

To obtain E_M we have

$$E_M = \frac{197.5}{.645} = 306 \text{ volts} \quad \#$$

Collecting our results, we have, at half power—

B.H.P. (motor)	= 335 H.P.
Kw. output (motor)	= 250 Kw.
E_L	= 527 volts
Motor eff.	= .93
E_T	= 10 volts

$E_{dc}M$	= 197.5 volts
$E_{dc}G$	= 135 volts
E_B	= 342.5 volts
E_M	= 306 volts
Gen. Loss	= 7.2 Kw.
I_{dc}	= 680 amps.
I_{ch}	= 520 amps.
I_B	= 840 amps.
$\alpha_1 G$	= 71.8 degrees
$\psi_1 G$	= 6.5 degrees
$\alpha_1 M$	= 30 degrees
$\psi_1 M$	= 30 degrees
α_{bat}	= 0 degrees
ψ_{bat}	= 16.5 degrees

To obtain the conditions with the motor stopped, we shall assume that when 2 I_{dc} reaches the value of 300 amperes the motor will stop. Also that we have a 5-volt drop through the motor and that the tube drop at this low rate is 9 volts. We shall further assume that the generator loss is equal to 4.9 Kw.

Then we have—

2 I_{dc} = 300 amps.

5 volts = drop through motor

9 volts = drop through motor tubes

9 volts = drop through Ch. tubes

Motor loss = $5 \times 300 = 1.5$ Kw.

Motor tube loss = $9 \times 300 = 2.7$ Kw.

Ch. tube loss = $9 \times I_{ch}$

Gen. Loss = 4.9

$$\text{Bat. Output} = \frac{I_B \times E_B}{1000} = 4.9 + 2.7 + 1.5 + \left[\frac{9 \times I_{ch}}{1000}\right]$$

$$\text{And } \frac{I_B \times 359}{1000} = 9.10 + 9\left(\frac{300 - I_B}{1000}\right)$$

$$359\ I_B = 11800 - 9\ I_B$$

$$I_B = 32 \text{ amps.} \quad \sharp$$

$$I_{ch} = 300 - 32 = 268 \text{ amps.} \quad \sharp$$

$$\text{And } E_B = 325 + \frac{1715 - 32}{50} = 359 \text{ volts.} \quad \sharp$$

$$E_{dc\ ch} = 359 + 9 = 368 \text{ volts.} \quad \sharp$$

$$368 = .670\ E_L$$

$$E_L = 550 \text{ volts.} \quad \sharp$$

$$\text{And } E_{dc\,G} = 359 - 9 - 5 = 345 \text{ volts.} \quad \sharp$$

$$\text{And } E_{dc\ G} = 345 = \frac{550 \times 3}{4\sqrt{\pi}} \frac{\cos(\alpha_{1G} - \psi_{1G}) + \cos\alpha_{1G}}{\sqrt{3/8 \sin 2\alpha_{1G} - 3/8 \sin(2\alpha_{1G} - 2\psi_{1G}) + \pi/2 - \frac{3\psi_{1G}}{4}}}$$

$$\text{And } \cos(\alpha_{1G} - \psi_{1G}) = \cos\alpha_{1G} + \frac{.134 \times 300}{1715} = \cos\alpha_{1G} + .023$$

$$\therefore \quad \alpha_{1G} = 25 \text{ degrees.}$$

$$\text{And } \psi_{1G} = 3.3 \text{ degrees.}$$

The preceding calculations are sufficient to cover practically any type of power circuit that will be encountered in actual operation.

SOUND AND VIBRATION IN MARINE GEARING.

By Austin Kuhns, Member.*

This article is the first of two to be written by Mr. Kuhns, in which he will deal with the problem of noise production and its control in gear manufacture. As the speed and power increases in marine and naval propulsion installations, the noise inherent to operation has tended to increase to a point where it must be definitely controlled if we are to avoid the unfortunate psychological and physiological effects which it wreaks on the plant operators when excessive.

Due to the acuteness of the problem brought about by the use of higher steam pressures, higher machinery speeds and more compact machinery layouts, it is receiving the most earnest study by all machinery designers, and Mr. Kuhns' long association with gear manufacture qualifies him to speak with authority on the elimination and reduction of noise in marine gears.

The present article states the problem and gives a general discussion of it. The second article, which will appear in the next issue of the Journal, *will discuss the experience which has been had in several specific instances to illustrate the progress which is being made in meeting the need for more effective control.*

DEFINITION.

Sound and vibration are cofunctional and inseparable. Vibration is a mechanical oscillation within a solid substance. Sound is a similar oscillation within a gas such as air, usually induced by vibration.

Noise is defined as those audible frequencies which subjectively affect the individual. The loudness of a noise is governed by the sensitivity of the ear to the different frequencies and sound energies. Annoyance is purely an individual matter, usually associated with high-frequencies and sudden or irregular clicks and thumps.

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MEASUREMENT.

The measurement of these values, therefore, becomes the first step in the study of this subject and in the formulation of the problems involved.

Annoyance, by its definition, defies expression in physical terms and is determined entirely by personal observation and experience. Annoyance to the point of distress is seldom encountered, but the degree to which it impairs operating efficiency is now realized to be a matter for serious consideration by marine designers.

Loudness, by definition, is measurable. The maximum sensitivity of the average ear is at frequencies between 2000 and 2500 cycles per second, with decreasing sensitivity at frequencies both above and below this zone. The value of the sound intensity at which loudness is introduced is a matter of individual opinion, being probably at 80 decibels for the average person, at which level the sensitivity characteristic of the ear to varying frequencies is approximately flat. The decibel scale, obviously fills the requirements for determining the loudness values.

In the measurement of noise, the decibel scale can be applied directly to all frequencies and intensities within the audible range. For the normal ear, these lie between 20 and 15,000 to 20,000 cycles per second and from 20 decibels upward.

Sound and vibration as purely physical functions are capable of scientific measurement and mathematical consideration. The energy is measured in decibels from which it can be converted into figures of acceleration, force exerted, vibration pressure or other physical units.

THE DECIBEL SCALE.

By definition, "The sound energy emitted by a machine is considered to be the average sound energy flux per unit area (normally one square centimeter) through an imaginary surface surrounding the machine and at a distance of one meter (approximately three feet) from the nearest point of the machine to each point of the imaginary surface, with the machine operating on the equivalent of a non-vibrating, non-resonant base, and in the equivalent of a non-reverberant room containing no other significant source of sound."

The unit of measurement is the decibel which can be expressed as

$$10 \log_{10} (P_1/P_0)$$

in which P_1 is equal to the power (or energy) of the sound or vibration measured and P_0 is that of a standard sound chosen as the zero of the decibel scale.

Unfortunately, hitherto, three different zeros have been used for scales of decibels, but at the present writing the zero of 1×10^{-16} watts per square centimeter is the accepted standard. For convenience in establishing this decibel scale on an understandable basis the following comparable average sound levels have been tabulated:

TABLE I.

	Decibels
Explosion of Krakatoa (doubtful)	about 190
Airplane engine and propeller	110 to 125
Boiler factory	105 to 115
Pneumatic riveter	100 to 110
Thunder	80 to 110
Niagara Falls	95
Average motor truck, on street	90
Average electric car, on street	80
Average horse-drawn vehicle, on street	80
Noisiest New York City street	81
Average city street	50 to 80
Average factory	50 to 90
Average city office	40 to 70
Piano practice	70 to 75
Vacuum cleaner	70
Average conversation	65 to 75
One typewriting machine in small office	40 to 45
Average city residence	30 to 55
Average country residence	30 to 40
Turning page of newspaper	30
Whispering	25 to 30
Open country at night	20 to 25
Noise of heartbeat, etc., of one person in sound-proof room	10 to 15

THE ACOUSTIMETER.

The results of experimental and development work have evolved several types of excellent instruments for decibel measurement, which satisfy the requirements of mechanical ruggedness, constant sensitivity and a regular response to different sound frequencies and the entire absence of internal disturbances.

The conventional design employs a condenser microphone with a mechanical diaphragm vibrating in an electrical field to produce a pattern of electrical impulses which are an image of the particular sound waves. These impulses are passed through vacuum tube amplifiers, transformer coupled, and the sound intensities are measured directly on the instrument scale. The frequency analysis is made by means of electric filters which separate the sound spectrum into a series of frequency bands for detailed study. For the finer and more accurate analyses of this spectrum the cathode-ray oscilloscope is necessary.

THE VIBRATION METER.

The design of the acoustimeter for sound is not suitable for measuring vibration. In fact experience has shown that two different devices are required for a complete and reliable analysis of vibration within a structure and, if possible, a noise survey as well to detect secondary vibrations or audible harmonics.

The simple vibration pick-up consists of a small probe, one end of which touches the surface of the solid while the other is attached to an electric coil which is set into vibration in a magnetic field. To overcome certain technical difficulties it is necessary to use a series of these pickups capable of responding to every kind of vibration.

For highly damped shock waves and frequencies below 25 cycles per second the output from the pick-up is connected to a cathode-ray oscilloscope. The screen of this instrument shows a reproduction of the wave forms from which it is usually quite easy to sort out and identify the vibration waves concerned.

For the higher frequencies the acoustimeter is used in conjunction with the magnetic pickup. By using this ahead of the oscilloscope, the picture of the low frequency vibrations will not become

confused by the high frequency vibrations existing at the same time.

The other methods of measuring vibration are based on the resonant reed and the seismograph respectively. The resonant reed is not sufficiently delicate nor does it respond to single shock wave or to impacts. The optical vibroscope is a variation of the seismograph. Its use requires three units, none of which respond to the higher frequencies, and the results must not be affected by the instruments themselves. If all of the characteristics of the vibration are not known, the conclusion reached from the use of the vibroscope can be highly misleading.

ROUTINE INSPECTION.

For the routine inspection of reduction gear units while in operation, relatively simple sound and vibration meters can be employed. Such a series of instruments can be permanently attached to the housing or used periodically as desired.

THE USE OF THE INSTRUMENTS.

With these measuring instruments the sounds and vibrations in the gear set occasionally can be traced to definite and recognizable imperfections in the unit, or to deviations from equilibrium conditions in the rotating members. Also amplification induced by the casing or reverberation in the surrounding atmosphere can be detected.

Unfortunately, the degree to which the development of sound and vibration measurement in gearing has progressed is still so far short of the goal that a complete and intelligent analysis of a specific set of conditions is impossible at the present time. The effects of noise, loudness, and annoyance, especially on shipboard, are forcing a rapid extension in a knowledge of the subject. It is now recognized that the noisiness which is so disagreeable can be the audible evidence of actual destructive vibration. Low frequency vibrations, which usually constitute a large part of the energy of machine vibrations, automatically are converted into sound waves of higher frequency.

THE NOISE PROBLEM.

From the practical standpoint and in the interest of constructive application of the available information, we are forced to recognize the present limitations and to consider by what means we can improve the general noise level of marine gears and correct the outstanding faults in the objectionable installations. We can consider ourselves fortunate when we are able to spot a definite defect and correct it immediately. Occasionally we can localize excessive noise and vibration, or detect frequencies which are induced from outside sources. Also the more familiar we become with the technique of sound and vibration measures, the more complete and useful will become our deductions from the figures obtained.

THE GENERAL NOISE LEVEL.

Sound and vibration are always present in gear sets of all classes. Perfection in design and manufacture, of course unobtainable, still would leave induced and sympathetic or resonant vibrations to contend with, which in themselves will naturally vary during the period of operating service. It is possible, however, to segregate or divide into classes the sources of sound and vibration excitation in gear units, and through intelligent application, when designing, to improve materially the operating condition of the propulsion unit. These classes are identified as primary, secondary, induced and resonant.

PRIMARY SOURCE OF EXCITATION.

The primary source of excitation in the reduction gear unit lies in the rotating members themselves (principally in the meshing gear teeth) and is amplified by the dynamic forces set up when the unit is in operation. Unfortunately, perfection in machine parts is as yet unobtainable and the effect of cumulative errors is unpredictable.

The gear teeth themselves are most difficult to manufacture accurately because of the many contour surfaces and the difficulty in producing exact duplication, and therefore, can be considered as the most likely, most prevalent and most serious source of excitation for gear noise. To attain perfection, the successive gear

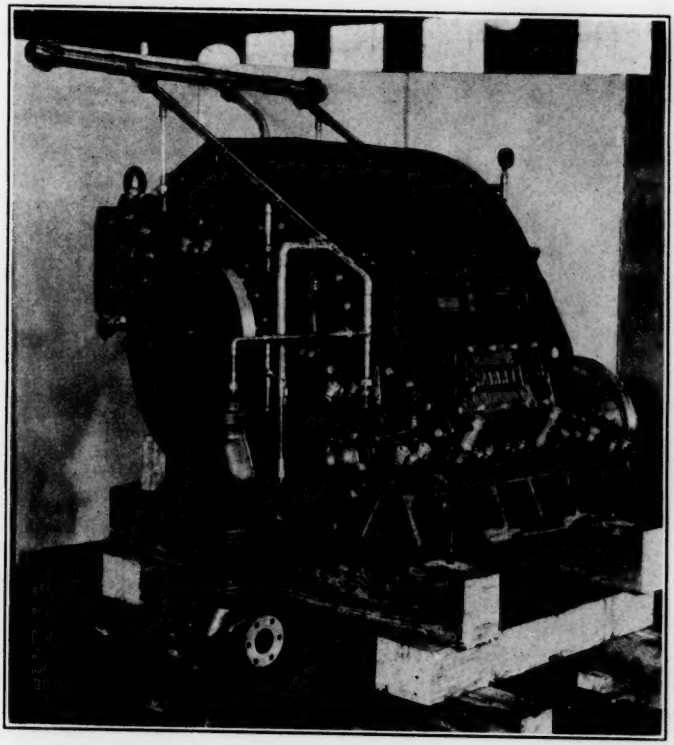


FIGURE 1.—A MODERN NAVAL GEAR ASSEMBLY. NOTE COMPACT ARRANGEMENT.

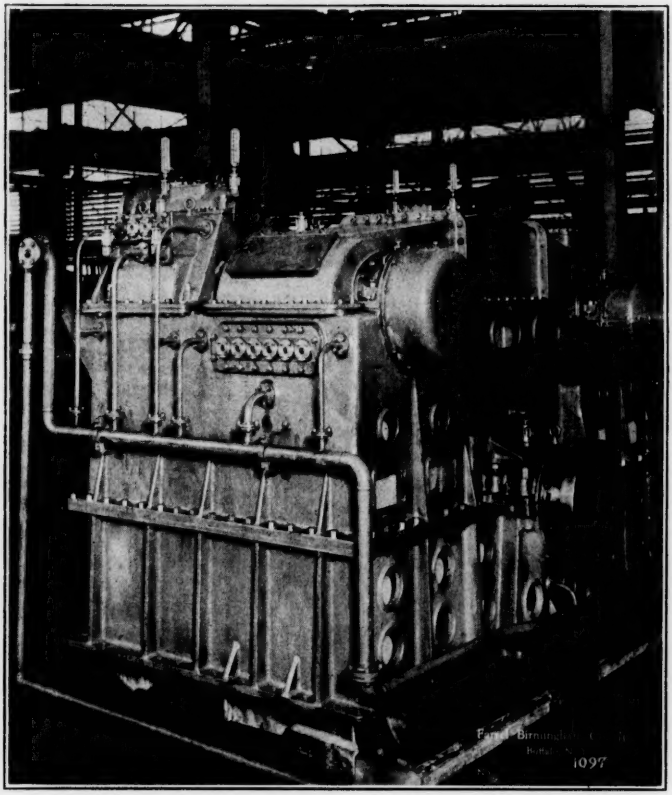


FIGURE 2.--A MODERN MARINE TANKER GEAR ASSEMBLY.

teeth around the periphery must have identical pressure angles, helical angles, pitch and spacing, and the contours of the tooth profiles must be precise, mathematically, with exactly the same characteristics. These same conditions must hold for each of the rotating gear members and furthermore must be absolutely the same for the different gears which are to operate in mesh together.

Experience has shown that comparatively slight differences in any one or all of these details can produce sufficient vibration under operating conditions to generate noise levels which are impossible to tolerate. Inaccuracies which are below the limit of measurement can produce objectionable results and, therefore, the practical production of an acceptable gear set becomes something of a matter of trial and error.

Furthermore the change in physical characteristics which the rotating member experiences under service conditions from such causes as expansion under operating heat, misalignment, damage from oil contamination and other causes, makes it impossible to guarantee continuance of the noise level established at the time of acceptance trials.

BENEFITS FROM PROPER GEAR TOOTH DESIGN.

Experience has shown that certain features can be embodied in the design of the gear teeth which will help materially to overcome the effects of imperfections in manufacture.

Of most importance is the use of high helical angles which automatically increase the overlap or number of teeth in contact and average the harmful effects from their inaccuracies. Theoretically the higher the helical angle the smoother the operation of the mating gear. Practically, certain difficulties arise in producing these higher angles so that actually they do not exceed 45 degrees. Also practice has established a minimum of approximately 30 degrees.

The effect of backlash on noise and vibration is little understood. Theoretically, the actual amount of backlash is of no consequence provided that there is a sufficient amount to permit free lubrication of the gear teeth. Practical experience also appears to substantiate this opinion.

The characteristic of creeping engagement with helical teeth is of real importance in the reduction of noise and vibration. This can be improved further by means of tip relief to permit smooth action of the meshing teeth and a preloading of the oil film at the line of engagement. Under these conditions impacts are eliminated.

The importance of pressure angle and depth of tooth to quiet operation in service is somewhat controversial. The longer tooth with the lower pressure angle and slight increase in line of contact is an advantage, but the manufacturing difficulties and the weaker tooth section seem to offset any gain.

In lubrication, the designer has his most effective tool for counteracting the errors in manufacture and installation. Gear teeth never operate with metal to metal contact. Consequently, the characteristics of the oil film between them govern the operating smoothness to a large extent. The capacity of the lubricating oil to distribute the load evenly over the theoretically rough metal tooth surface acts to prevent the excitation of vibration and noise. The dash-pot effect operates to dampen frequencies already set up.

MANUFACTURING DIFFICULTIES.

The greater the number of successive machining operations, the more difficult it becomes to approach perfection. In the manufacture of the rotating gear member, we are confronted not only with ordinary machining operations such as turning, boring, grinding and the like, but we are required to form gear teeth in the peripheral rim as well.

The accuracy of boring and turning operations depends upon the quality and constancy of the material, the proportions of the particular gear member, the accuracy of the machine tools and other shop facilities, and the ability of the operators to handle them. It is customary today to grind and polish the shaft journals and to permanently mount the gear wheels before the cutting operation. As a refinement, the gear wheels and pinions are finished by grinding them while they are supported on their shaft journals.

With equipment available at this time it is possible to hold the average journal diameter to .0004 inch cumulative error, and the

pinion body to .0005 inch. The accuracy of the finished gear blank is of course better with a small diameter, but even the largest wheel can be machined to a tolerance in diameter not to exceed .001 inch. The trueness of the faces or sides of the pinions and wheels should be held to like accuracy.

GEAR CUTTING AND FINISHING.

The operation of gear cutting quite correctly has been held responsible for the major troubles in marine and high-speed gear units and, in consequence, the different methods of generating the gear teeth have become the subject of considerable controversial discussion. In recognition of the fact that the subject of this paper is limited to a discussion of gear noise, we consider that it does not fall within its scope to discuss the different gear cutting processes or to express an opinion regarding their relative merits. It is most important, however, to point out that if the desired results are to be obtained and the inaccuracies in helical angles, pressure angles, pitch, spacing, smoothness and perfection of tooth profile and tip relief are to be reduced to a minimum, the only gear generating machines which can be used, are the best obtainable, designed along proven lines, and of a weight and size sufficient to avoid the pitfalls of elasticity, distortion, twist, and weaving in the cutting elements.

Whether or not the gear members, after removal from the machines, are subject to further finishing and refining processes, often grouped under the poorly selected name "lapping," is subject to a wide difference of opinion and so does not lie within the scope of this paper. We must point out, however, that if the accuracy and condition of a tooth surface which results directly from the gear cutting operation compares favorably with that obtained from lapping or finishing processes, the gear members when removed from the cutting machine, can be considered finished.

True lapping or finishing consists of a series of operations, wherein the gear members are first lapped with an accepted abrasive compound, next burnished with special lubricating oil under pressure and finally polished with light lubricating oil under a combination of pressure load and velocity. The first and second steps are carried out on each gear member successively. The polishing operation requires the several gear members to be in mesh and serves

also to indicate the alignment and general operating characteristics of the gear unit before it is installed and placed in service.

Experience has shown that if when mating gear members are set up under load in flexible cast frames suitably fitted with ten-thousandths dial indicators, the total cumulative movement does not exceed .001 inch, the gear unit when in service will be acceptable. This method of inspection discloses inaccuracies in pressure angle, helical angle and tooth spacing, and inequalities or roughness in tooth surfaces which cannot be detected or measured in any other way.

THE SECONDARY SOURCES OF EXCITATION.

The secondary sources of excitation of gear noise and vibration come from faulty operation caused by imperfect assembly of the gear members either through poor design, inferior machine work, careless direction and alignment, unbalanced expansion at operating temperatures or, finally, from critical vibrations within the unit itself. Inaccuracies in journals or bearings, eccentricity of gear blanks and other similar defects, contribute through their defects on the meshing of the gear teeth sufficient excitation to cause vibration and excessive noise.

An out of balance condition, especially dynamic, need not be large in high speed gearing to produce excessive vibration and dangerous operating conditions. It is not enough for the smoothest operation in service to balance statically and dynamically the high speed rotating elements, but the slower turning gear members should be balanced as well. To avoid distortion in the finished gears these rotating parts must be balanced before the teeth are finished and rebalanced after cutting to compensate for any change introduced by this operation.

Serious misalignments can be induced through insufficient rigidity of the gear casing itself or through the use of faulty foundation supports. The fact that the marine propulsion unit usually includes the main propeller shaft thrust bearing further complicates this problem of rigidity and alignment. Although lightweight design is a great advantage, too much emphasis on this feature can produce a condition of misalignment which cannot be rectified and which can be the cause of vibration and noise correctable only by the installation of new and heavier housings.

Critical vibrations even of the minor amplitudes, can introduce variations in depth of mesh and angular velocity of the gear teeth which will cause excessive noise. As with misalignment and other secondary causes critical vibrations become serious, not only in their acoustic effects, but in the abuse which these conditions can impose on all of the component parts of the unit. If allowed to continue, fatigue, crystallization, and failure, not only of the gear teeth themselves, but of the shafts can be expected.

INDUCED VIBRATIONS.

The third source of excitation has been defined as the induced type and includes all of those changes in alignment and velocity which result from the operating characteristics of the driving and driven elements. Since, for successful operation there must be backlash or actual physical clearance between the gear teeth, the effects of changes in angular velocity become automatically complicated and magnified. Vibrations in the driving element caused by operating at critical speed or induced from torque variation or inaccuracies of manufacture, are transmitted directly to the reduction unit and to the line of gear mesh, where they excite further vibrations which in turn are converted into noise. The same situation applies to similar vibrations generated in the driven element.

If the entire installation is engineered with thoroughness and completeness and a mathematical study is made by a competent authority, the amount of induced vibration from criticals, torsionals or other sources in the connected machinery becomes negligible. The benefit which follows a reduction of these vibrations is expressed not only in noise diminution, but in relief of the stressed material itself.

RESONANCE.

The final source of excitation is resonance, either in the gear unit assembly, or between it and unrelated or connected machinery elsewhere in the hull.

Resonance to a moderate degree in the gear set casing is not unusual. The characteristics of rolled steel plate make it especially susceptible to the excitation of sympathetic vibration unless special care and attention is given to details in the design of the welded structure to counteract this condition. Shaft steel and the cast

irons, on the other hand, possess definite dampening qualities caused by the natural uneven metal section and the properties of the grain structure.

Sympathetic vibration between the unit and unrelated or connected machinery seldom are the cause of objectionable noise and vibration. Frequently, however, it is noticed that gear noises vary not only with speed, but also with the different loads and the operating conditions of other equipment in the same compartment or in the hull. The investigation into the causes of such sympathetic vibrations is highly technical and at best relies heavily on a program of trial and error.

VALUE OF THIS STUDY.

The obvious purpose of any study of sound and vibration in marine gearing is to evolve ways and means to improve its operating characteristics and to correct specific troubles with individual units. As we have seen, delicate and accurate instruments are available for the scientific analysis of sound and vibration waves. Added to these, we have the torsiograph to disclose torsional oscillations in the rotating members.

We have also seen that the pattern of vibration and sound waves in the reduction gear unit actually become so complex and extend over such a wide range of frequencies, and can be confused to such a marked degree by resonance and reverberation, that a comprehensive mathematical analysis is impossible. We do find, however, that an intelligent diagnosis of the general physical conditions of a gear unit can be made, and that we can use this as a basis for continued improvement in our design, manufacture and operating procedure.

METHODS FOR IMPROVEMENT.

Obviously, any general attempt to better the performance of marine gearing must commence with the correction as far as possible of the several sources of excitation of the sound and vibration waves.

IMPROVEMENT IN ROTATING GEAR MEMBERS.

Improvement in the manufacture of the rotating elements correctly receives our first attention. Every gear maker spends huge sums yearly for new and better shop equipment, and for special

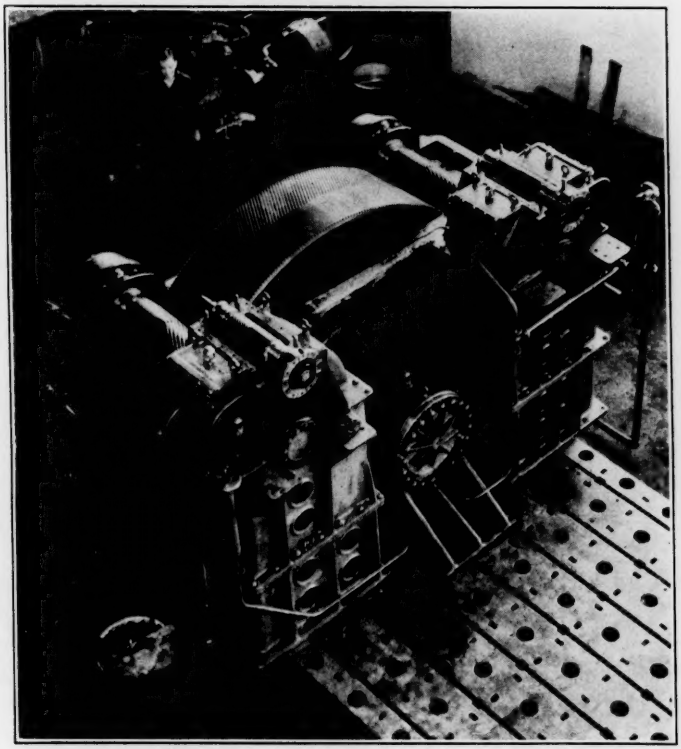


FIGURE 3.—A MODERN MARINE TANKER GEAR ASSEMBLY. TOP CASING REMOVED.

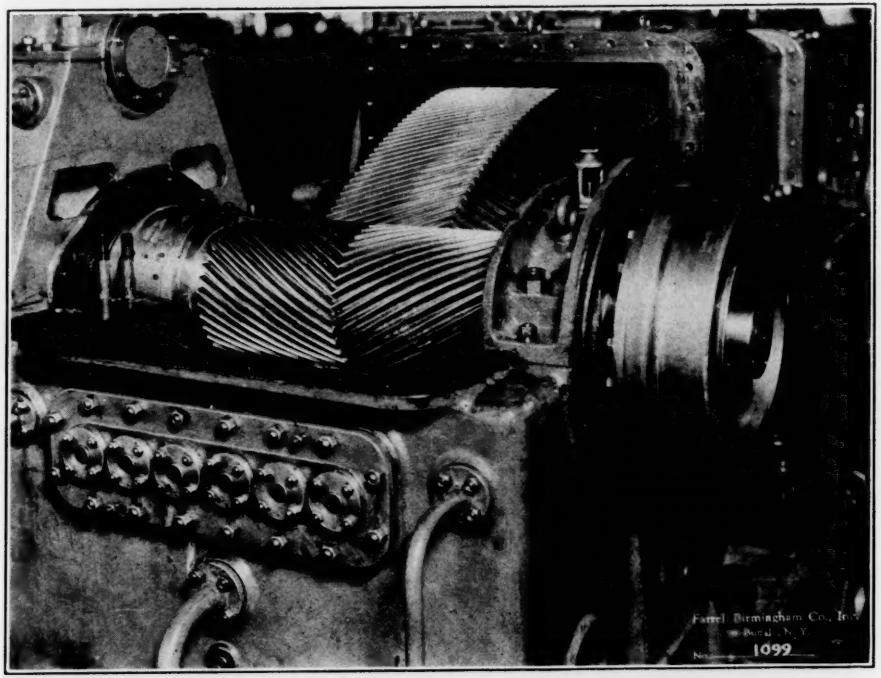


FIGURE 4.—A MODERN MARINE TANKER GEAR ASSEMBLY. CLOSE UP TO SHOW BULL AND PINION MESH.

fixtures and tools to reduce the inaccuracies and imperfections in his product. Real ingenuity is constantly exercised in the development of gauging and other instruments for physical measurements. Marked progress is being made in methods for finishing and polishing the gear tooth surfaces and in the accuracy of balance of the rotating members. The pace of future improvement will depend largely upon the enthusiasm and support extended to the manufacturers by the users of fine gears.

IMPROVEMENTS IN GENERAL MECHANICAL DESIGN.

For the reduction of the secondary source of excitation, the manufacturer must rely again upon the stimulating interest and exact demands of the users. Sound engineering, good mechanical design and a competent shop and field organization can reduce the magnitude of these secondary excitations to a negligible amount. This goal is not easy to attain. Extended experience and the resources of research and experimentation as well as the availability of the best obtainable engineering department are required. The resulting gear units will be costly, but not too much so when the vital importance of continued operation of the gear units to the profitable existence of the ship itself, is clearly realized.

REDUCING INDUCED VIBRATION.

Too little attention has so far been given to the operating characteristics of the driving element and of the driven propeller with shaft and to the effect of these on the reduction gear unit. Only by considering the propulsion machinery as a single integral unit can the possibility of induced vibrations in the gear set be eliminated. Actual experience has proved that there are no difficulties or questionable expenses involved in a thorough and complete mathematical study of the propulsion unit as a whole and specific installations have demonstrated that objectionable conditions can be predetermined and the danger of induced vibration removed. This is especially true where gear units are used between one or more Diesel engines and the propeller shaft where the effects of the reciprocating motion in the high speed engine and the characteristics of the crank shaft and other parts produce a complex pattern of stress application.

Show Bull and Pinion Mesh.

RESONANCE AND REVERBERATION.

The possibility of reducing resonance and sympathetic vibrations offers considerable latitude for investigation and the exercise of ingenuity. Since every installation is different and dependent almost entirely on the architectural arrangement, the problem on each ship must be considered as a separate piece of research work.

The first step is the correction of any conditions of resonance within the gear unit itself. From the practical standpoint, we are limited to actual changes in the casing itself, such as dampening strips or buttons, added reinforcing ribs, thicker or thinner metal sections, altered lightening holes, tierods, spacers, or the like. It is possible to a certain extent to predict the results from such changes. In the main however, this becomes a matter of trial and error.

To correct sympathetic vibrations from connected or unrelated machinery is far more difficult and requires an extensive study into the operating characteristics of every piece of equipment in the compartment and hull. With the use of the acoustimeter, vibration meter, the cathode-ray or oscilloscope, the torsiograph, and by the orderly process of elimination, the offending feature of the machinery probably can be identified. Correction obviously requires changes in material or method.

Reverberation, and to a certain extent sympathetic vibrations, can be controlled and often reduced with the intelligent use of sound insulation and sound absorbing material. Unfortunately, unless these various products are used sensibly, the results can be very disappointing.

Insulation of the main propulsion gear unit by inserting some dampening material between its supporting flanges and its foundation, is a highly debatable question, mainly because of the concurrent requirement that the thrust from the propeller must be taken through these layers of insulating material, and also because of the fact that the shaft connections through the driving element and to the propeller, contain no insulating or dampening feature. To coat the exterior of the gear set casing offers a certain possibility of relief, but such a covering, of course, is equally effective in reducing the highly desirable radiation of the operating heat.

Therefore, we believe at this time that the intelligent use of sound and vibration absorbing materials on the walls of the containing compartment and on machinery which cannot be adversely affected by it, becomes the most promising method for improving the sound level in the ship.

MOLDS FOR PHENOL RESINOIDS.

By T. E. Cassey, Member.*

In the February, 1937, issue of the Journal, *Mr. Cassey presented an article entitled "The Problem of the Phenol Resinoids," which did much to clear the air between the Navy Department and the manufacturers in the much discussed question of procurement and supply of phenolic materials. It will be recalled that at that time a second article was promised which would cover mold manufacture, design, and technique, thus forming a basis for a correlative understanding in an equally difficult and involved matter of proper supply. It will probably be apparent at once that any difficulty or misunderstanding which may now exist between buyer and vendor, centers around the fact that it is perfectly feasible to manufacture a good mold very cheaply which will serve all purposes in the production of a limited number of parts on a particular order. But the Government always wants more than this, and demands a mold which can be used again and again, and by many different manufacturers, and for this purpose, the cheap mold will not serve, and resort must be had to a standardized, rugged, and high-grade article which has been thought out, designed and built against a "long haul." Mr. Cassey deals with the variations in type which appear to be necessary today, together with little tricks of design detail which have been noted as the result of a wide experience in purchase.*

In a discussion of the problem of Phenol Resinoids, in the February (1937) number of the Journal of the American Society of Naval Engineers, reference was made, somewhat briefly, to the question of molds and their relation to the production of molded phenol resinoids.

* Chief Electrical Draftsman, Bureau of Engineering, Navy Department, Washington, D. C.

The popular conception of the word mold calls to mind, immediately, a device in which some liquid or plastic material is poured, to obtain some desired shape. So numerous are the applications of molds, of this category, to both industrial and household uses that it would be indeed difficult to find a person who could not describe, with little effort, some sort of mold with which he is familiar. In no application of the mold to industry do we find any particular difficulty of understanding or complexity of design until we come to the phenol resinoid molding industry. Unlike all other industries the mold, as applied to the production of molded phenol resinoids, is so intimately concerned with the basic raw materials with which it is used that it is difficult to disassociate the two problems even for the purpose of technical discussion.

The general meaning of the word "mold" and its variety of applications contributes somewhat to the difficulty of visualizing this equipment as applied to the phenolic molding industry. Webster's dictionary defines the word "mold" as "The matrix or cavity in which anything is shaped and from which it takes its form; also the body or mass containing the cavity." Many persons apply the word "die" to the phenolic molding "device." The word "die" is defined as "A metal block or plate (or pair) shaped to give a certain desired form to . . . by pressure." Because neither of these definitions is entirely applicable, it appears desirable to establish a more exact definition of the term "mold" as applied particularly to molded phenol resinoids. The following definition appears fitting:

"The cavity (or cavities) in which the material takes its form: also the body or mass containing the cavity (or cavities); involving a predetermined pressure and temperature for a stated duration of time, and including the parts necessary to eject the finished piece."

With this definition in mind, it is desirable, as the next step, to establish some suitable classification nomenclature which will make possible the definite identification of the various principles embodied in phenolic mold designs. The lack of this classification data has been one of the contributing factors responsible for the various diverse "leads" which have been followed in the last few years in an effort to "work out" the advantages and deficiencies of

various mold designs advocated by manufacturers, as applied to the production of Navy designed pieces.

Some indication was given of the various types of molds used commercially, in the February issue of the JOURNAL, already referred to. The classification data, however, now presented is more complete and is believed to be the first attempt, industrially or otherwise, to establish a comprehensive, workable basis for the identification of these molds which will be of real value in the future discussion of this class of equipment.

With reference to the principles involved in the construction and operation of molds for phenolic resinoids, a complete classification is given as follows:

Construction
- Unit
- Built up
- Split follower
- Ring follower
- Split cavity
- Floating chase

Heat Transfer
- Direct
- Indirect

Method of Handling
- Hand
- Semi-automatic

Number of Cavities
- Single
- Multiple

Physical Action
- Flash
- Positive
- Extrusion

Loading
- Cavity
- Filler plate

Closure
- Complete
- Spacer fork
- Spring box
- Spacer fork

Ejecting
- Knockout plate
- Stripper plate
- Automatic

Raw Material
- Bulk
- Preform

In elaboration of the above we will first consider the principles under the term "Construction." As an explanation of these classifications involves mention of that part of the mold known as the "chase," it is well to define this piece at the outset as the "central member of the mold which bears the molding cavity or cavities."

The mold cavity, sometimes called the "die," is that part of the chase in which the molded piece is formed. The cavity or cavities are either cut directly into the chase or formed within a plug or plugs which in turn are fitted in the chase.

Where the cavity or cavities are cut directly in the chase itself, the mold is termed a "UNIT CONSTRUCTION" mold, and when formed by the use of separate plugs, fitted in the chase, the mold is called a "BUILT UP CONSTRUCTION" mold.

The built up construction mold may be of four different forms. In one form the cavity is machined in a cylindrical plug and the plug fitted tightly into a recess cut in the chase to receive it. In the second form, probably the most common, a through hole is cut in the chase of the size and shape of the molded piece; a bottom plug is fitted in this hole flush with the bottom of the chase, forming the bottom of the cavity. The third form is the split follower type wherein the separate cavity plug or plugs are made in two or more pieces. This type of mold is designed for molding of pieces having undercuts or projections, which would make their removal from the simple "solid cavity" mold impossible. The remaining

form is the ring follower mold, used generally in the case of circular pieces having molded threads, to permit these pieces being ejected from the mold.

With respect to heat transfer, the second classification, molds may be classified as INDIRECT HEATED and DIRECT HEATED.

When molds are heated by being placed between hot press platens, they are indirectly heated. When the mold itself is channeled for steam and water circulation, and provided with suitable piping connections therefor, it is considered as directly heated.

The application of heat to the mold is of great importance. In order to insure uniform curing of the piece, the design of the mold in so far as is practicable should be such that all parts of the mold cavity will receive a uniform heat. Commensurate with other design requirements, the mold should be of minimum mass and the heat should be applied as directly as possible to the mold. Where steam heating is used, no other accessories for heating, such as electric heating elements and gas burners, should be utilized.

The ideal mold from the standpoint of heating is the direct heated mold in which the steam circulates in the mold block itself, close to the cavities. In a mold of this type the heat loss through radiation is far less than where the mold is heated from contact with platens. Also the transfer of heat units to the phenolic material is quicker and more efficient for there is a short direct path through the metal. Care should be taken in this design of mold to see that all parts of the mold will be heated uniformly and that there will be sufficient volume of steam going through the mold to supply the proper amount of heat.

Direct heated molds are confined to those of the semi-automatic type and are usually press mounted due to the required steam and water connections. In the case of some semi-automatic molds, where the dies are backed by steam heater-plates, forming an integral part of the mold, the application of heat is indirect, from heater-plate to mold cavity; however, inasmuch as the heat-transfer surfaces remain in their original condition, the heating characteristics approach those of direct heated molds, and such molds may be so considered.

Hand molds, by reason of their "hand handling," are necessarily of the indirect heated type. With this type of mold, since the heat is transferred from the platen to the mold, it is very important that there be a close contact between the mold and the platen. In order to keep the heat losses to a minimum, it is important that the outside surfaces of platens and molds be kept smooth and clean. Platens dented and containing spots of cured material cause a poor contact and greatly increase the heat loss. These platens often become partly clogged from scale and sediment deposited and consequently cut down on the volume of the steam flow. They should frequently be checked and cleaned or replaced when necessary. Hand molds tend to cool off considerably while they are out of the press and it is essential that they be replaced in the press as quickly as possible to avoid too great a loss of heat. For the most efficient operation it is good practice to keep them on an auxiliary hot platen while they are being loaded.

With respect to the Method of Handling (loading and pulling), the third classification, molds may be classified as HAND or SEMI-AUTOMATIC.

Hand molds may be defined as those molds which are moved in and out of the press manually by the operator. Loading the mold charge and placing the inserts is done on the bench. The opening of the mold and ejection of the piece is accomplished in a separate stripping press, or on a bench.

Weight and size are the deciding factors in the use of hand molds. A maximum weight of 100 pounds and a size limitation of 16 inches $\times$ 16 inches may be generally accepted as workable limits for good hand mold operation.

When the mold mass is such as to offer difficulties in "hand" handling, chain falls, slides, or other means as desired by the manufacturer are used. In such cases, however, the use of the hand mold in comparison with the semi-automatic mold, where increased difficulties in handling are involved, resolves itself into a question of economic production.

In general, hand molds are to be preferred for the production of Navy designs of molded parts in spite of the advantages which semi-automatic molds may hold for high speed and quantity pro-

duction in commercial industry. Hand molds have advantages of simplicity and low cost, they offer greater facility in placing inserts, and greater economy of maintenance. Hand molds are also more suitable for a relatively small and periodic production, and moreover, provide a wider measure of competition for Navy orders, as molders can make use of hand molds without alterations to their existing plant equipment. Hand molds furthermore lend themselves to greater ease in shipment to and from the custodian Navy Yard.

As compared to these advantages, hand molds have the disadvantages of low rate of production, high operating cost, and of poor economy due to indirect heating. In addition, the size of hand molds is limited by weight considerations, as previously stated.

Semi-automatic molds are defined as those molds which are rigidly attached to the molding press. Loading is done with the mold in position and the pieces are automatically ejected when the press opens. With this type of equipment, the speed of the molding cycle depends largely upon the plastic used and is relatively independent of the style of mold; therefore, the total molding cycle may be appreciably shortened only by increasing the rapidity of charging the mold and of the ejection of the finished parts. When quick-curing materials are used, semi-automatic molds are inherently conducive to high-speed production. A more uniform cure can usually be expected with semi-automatic mold equipments due to better heating characteristics; therefore, this type of mold is more practicable for molding plastics requiring long-time curing.

For Navy design molded pieces where the molded part is to be produced in such large numbers as to justify the procurement of the more expensive and complicated semi-automatic molds, care should be taken that the successful bidder on the first order does not thereby obtain a virtual monopoly of subsequent business on this particular part. If and when semi-automatic type molds are used, these molds should be required to be completely self-contained.

Based on the number of cavities, the fourth classification, molds are classified as SINGLE CAVITY or MULTIPLE CAVITY.

It is in the matter of cavities that the greatest difference lies between the molds used for industrial production and those required for Naval production.

A few general rules may be laid down which govern the applications of single and multiple cavity molds for industrial production. Single cavity molds are employed generally under the following conditions:

In cases where a small quantity of relatively complicated parts are required.

When the molding is of such size as to require the total capacity of a press.

Where rate of production is of secondary importance and the volume of production can absorb only the minimum mold cost.

Where the raw material is not free flowing, particularly of the high-impact type.

Where the placing of metal inserts within the cavity is of sufficient complication to prohibit the economical use of multiple cavities.

As a test mold before constructing a multiple-cavity mold, to prove the design.

The converse of the above limitations as applied to single cavity molds, in general may be accepted for the application of multiple cavity molds. More specifically, multiple cavity molds are ordinarily high-production equipment; but due to their higher initial cost, their use is justified only where a large volume of production is to be expected. It is a great temptation to put a number of cavities in one mold in order to save on the direct labor cost, but this hoped for saving can be lost in so many other ways that it is highly important to consider the matter carefully. The selection of the number of cavities for a multiple cavity mold involves the careful study of the procurement factor involved in order to obtain the proper economic balance.

The present trend is away from large multiple-cavity molds, most molders preferring to use more molds, each having a small number of cavities. This is particularly true of parts of moderate complexity having inserts; but to some extent applies to even simple molded shapes without inserts. One reason for this is, of course, the expense involved in making large multiple-cavity molds which require large presses to mount them. Damage to one cavity is almost certain to occur sooner or later, and the damage then is likely to become progressive until finally the mold has to be scrapped.

A multiplicity of smaller molds permits greater flexibility of operation in the plant, and enables the molder better to match production to demand while maintaining an even flow of work. Damage to one cavity of one mold can be repaired without greatly slowing up or entirely stopping production.

An important factor which should be considered in determining the number of cavities to build into a mold, is the number and type of inserts required. Where two molds are used, the time required to load the inserts should not exceed the molding time required, but should approximate it as closely as possible so that while one mold is in the press the operator can load and prepare the other. Where more than two molds are being used, loading time should be so related to curing and molding time that the operator can run practically a continuous process.

In the case of the production of pieces for Naval installations the stringent requirements for control of the density of the product point toward the minimum number of cavities, as the chances of keeping the grade of the piece to a high quality is increased with molds of few cavities.

For high impact materials, for Naval production, experience has shown conclusively that molds should be single cavity. High impact materials have a large bulk factor, making the loading of the cavities more difficult. With the possibility existing of even a slight differential in the amount of material in the cavities of a multi-cavity mold, it has been found that the pressures in the cavities will vary, resulting in one cavity "taking the load."

For other than high-impact materials, it is safe to state that the employment of single versus multiple cavity molds, for Naval production, will follow somewhat closely the rules for selection set forth above for industrial uses.

As regards the basic principle involved in their compression, or what may be termed the physical action of the plastic during molding operation, which is the fifth classification, molds are divided into types as follows:

Flash
Positive
Extrusion

Considering, in conjunction with the "physical action classification," the next three classifications: "Loading Principle," "Ejecting Principle" and "Raw Material," classifications may be established as follows:

As concerns the loading principle, molds involved are either
Cavity Loaded or
Filler Plate Loaded;

second, as to their closure molds, are
Complete
Spacer Fork
Spacer Fork, Spring Box;

and, third, with respect to their ejecting principle, molds are
Knockout Plate
Stripper Plate
Automatic.

Finally, as regards the form of the raw material used, molds may be classified as
Bulk or
Preformed.

The classifications of "Loading," "Closure," "Ejecting" and "Raw Material" are all intimately connected with the "Physical Action Classification" and can be more clearly discussed as they become applicable under the various types of molds coming under that heading.

Considering first, under the "Physical Action" classification, the flash molds, we find this group further subdivided into the following subtypes:

Overflow
Subcavity.

In general, flash molds differ basicly from all other forms of molds in that there is no telescoping of the force plug within the molding cavity, thus the molding cavity is the exact depth of the finished part required.

FLASH OVERFLOW MOLDS.

The flash overflow mold is the simplest form of flash mold. In this type the corresponding surfaces of the force plate and the

chase, normally in contact when the mold is closed, are cut away or relieved to form an overflow space or "flash chamber" into which excess material may escape during closure of the mold. It is from this "flash chamber" that the mold derives its name. A narrow strip of contact surface, extending around the periphery of the mold cavity, is retained between the mold cavity and the flash chamber and it is over this "flash" or "cutoff" surface that the excess material flows. The final closure of the mold takes place upon a thin fin of polymerized material thus formed, the halves of the mold remaining separated by the distance equal to the thickness of this fin. The thickness of the fin depends upon the mold pressure, the mold charge and the width of the flash surface. Care must be exercised in this type of mold, to see that the flash area (flash cutoff) is of proper size. Too great an area will produce a heavy fin which is difficult to remove; too little area will cause crushing, due to extreme pressure on the flash area, and will present insufficient resistance to the molding material, producing too low a pressure on the charge for proper molding. In general, the flash surface should be from 1/8 inch to 3/16 inch wide.

The flash ridge serves as a means of obtaining back pressure on the molded piece as the material polymerizes during the flow. The requirements for this back pressure and for sufficient strength to withstand mechanical damage by the molding pressures, determine the safe minimum width of the flash ridge. Fin thickness of 0.005 inch to 0.008 inch is desirable, and can be obtained by properly proportioning the cavity and cutoff area when using ordinary molding pressures.

A mold charge containing an excess of material, is used with the flash overflow mold, the additional material being necessary to insure sufficient back pressure to form the molded piece. Where the design of the molded piece permits, the mold charge should be of bulk material, otherwise preforms will be used. When bulk material is used, the material should be heaped up in the cavity with the apex of the pile near the cavity center.

Ejection of the finished pieces for the flash overflow mold is usually accomplished by the use of a knockout plate.

In view of their relative simplicity flash overflow molds can be built more cheaply than all other mold types, but as is usually the

case with the "cheapest machine," have certain disadvantages. These molds require heavier charges due to the corresponding loss in "flash," and also involve a cleaning operation for the removal of this flash fin. In addition, the pieces produced in flash molds are quite apt to be of less density than pieces produced in other forms of molds, and also their density will not show as good uniformity. Pieces, moreover, will vary slightly in dimension, in the direction of the travel of the force. This is due to the fact that the final effective pressure is exerted directly on the fin of material which has flowed over the flash ridge. Inherently the thickness of this fin will vary with different charges of the same mold and also differ with the cavity of a multiple cavity mold.

Flash overflow molds are in general suitable for the production of relatively simple shapes, and where strength and uniformity are not primary considerations; also where the removal of the fin will not damage the molded surface of the finished piece or such removal operation does not constitute a major item in the cost of the product.

For Naval production this type of mold should be limited to the production of simple forms of insulating bases, bushings, handles, etc., of thin sections, and using wood-flour filler material. The use of fabric base materials with flash molds should not, in general, be permitted. For Navy designs, it has been determined that the maximum permissible flash thickness should be such that the finish edge of the piece, after removal of the flash (chamfer), shall not exceed 1/32 inch. This chamfered edge is required to be ground smooth and buffed.

The flash overflow mold may have incorporated in its complete classification any of the forms listed under the principles of "Loading," "Closure," "Ejecting," and "Raw Material." Figure I shows the essential features of a Flash Overflow Mold; Cavity Loaded, Complete Closure, Knockout Plate, either Bulk or Preformed Material, as applied to Naval production. The mold illustrated is also of the following additional classification: Unit Construction, Indirect Heated, Hand, Single Cavity.

As will be seen from this sketch, the principal parts of the mold are the top or force plate, the force plug, the bottom plate, and the chase. The chase rests on the bottom plate being positioned by

four large dowels known as guide pins. These guide pins are shown shouldered and riveted to the bottom plate. Riveted to the force plate in like manner are four similar guide pins from which the force plate receives its proper alignment in its downward travel under pressure from the press. Directly in line with the guide pin holes in the chase and in the opposing plates from the top and bottom sets of guide pins, holes are drilled of the same diameter to avoid any possibility of obstructions interfering with the travel of the guide pins as the mold closes.

It will be noted that the guide pins, both in the force plate and the bottom plate are located so as to prevent interchangeability of the plates and also to prevent the positions of those plates being reversed.

Secured to the force plate by cap screws is the force plug which forms the upper face of the molded piece. The force plug is also doweled in position. Due to the construction of the flash overflow mold, where the force plate comes in contact with the top of the cavity in the chase, the force plug is, of necessity, recessed to its full thickness in the force plate. For ease of manufacture, the force plug generally is machined to the more complicated shape of the piece.

For forming holes in the molded piece, hole forming pins are provided. Figure I shows these pins shouldered and riveted in place. Figure I does not show any inserts, but where inserts are required, these inserts are positioned by holding pins somewhat similar to the hole forming pins.

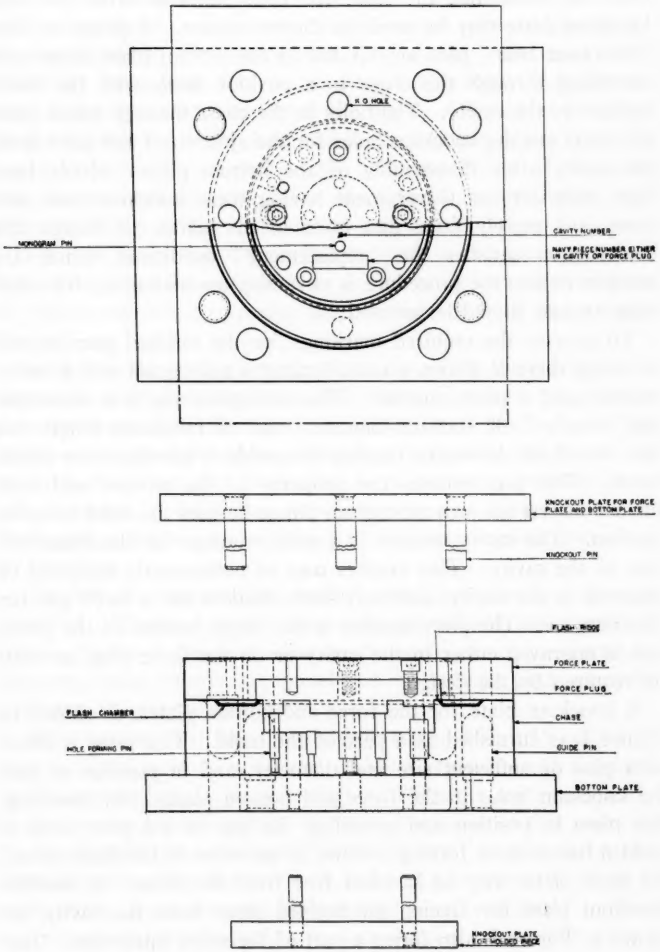
Hole forming and insert holding pins should be held sufficiently secure in the mold to withstand molding pressure, and should be easily replaceable. This is rather a difficult requirement and a matter of much controversy. Further discussion of this point will be found under the heading of " Detail Design " later in this article. In some cases manufacturers, in order not to secure pins to the mold plates, use a separate plate known as a " pin plate." This is not objectionable, except for the introduction of another piece to the mold assembly.

Holes are shown in the force and bottom plates, which are known as knockout holes, these holes being for the purpose of freeing these plates from the chase after the molding operation. These

holes are positioned the same for both plates in order that one knockout plate may be used for the two plates. A group of four "knockout hole" pins are riveted to the bottom plate, these pins extending through the chase to a position flush with the lower surface of the cavity. The holes in the chase through which these pins pass are the knockout holes for the ejection of the piece from the cavity after disassembly of the bottom plate. Molds have been built for the Government having these knockout hole pins loose, and merely fitted into holes in the plates. Although this construction satisfies the "replacement" conditions, which the molders desire, the procedure is obviously unsatisfactory for other reasons, and should be avoided.

To provide the required markings for the finished piece, it will be noted there is shown a manufacturer's monogram pin, a cavity number and a piece number. The monogram pin is a removable pin, usually 7/32 inch in diameter, and of minimum length not less than three diameters, bearing the molder's monogram or trademark. This pin remains the property of the molder and each molder inserts his own monogram pin as he uses the mold for production. The cavity number is a serial number for the identification of the cavity. This number may be permanently engraved or stamped in the cavity, although some molders use a mold pin for this purpose. The piece number is the Navy number of the piece, and is engraved either in the cavity or on the force plug, as may be required by the design.

A knockout plate for the force and bottom plates, as shown in Figure I, is furnished as a part of the mold. This plate is fitted with pins of sufficient size and diameter, and in position to suit the knockout holes in the force and bottom plates. By inserting this plate in position and pounding the top of the plate with a padded hammer, or forcing it down in an arbor or knockout press, the mold plates may be knocked free from the chase. A smaller knockout plate for freeing the molded piece from the cavity, as shown in Figure I, also forms a part of the mold equipment. The pins in this knockout plate fit the knockout holes in the bottom of the chase previously described. Some manufacturers use what is termed a "progressive knockout plate" which combines all necessary knockout facilities of the mold in one plate.



UNIT CONSTRUCTION, INDIRECT HEATED, HAND, SINGLE CAVITY, FLASH OVERFLOW, CAVITY LOADED, COMPLETE CLOSURE, KNOCKOUT PLATE EJECTING MOLD

FIG. 1

Four flash overflow type molds are now being used for the production of Navy designs of molded phenolic pieces, the designs of these pieces being relatively simple. Two of these molds are single cavity, hand type and the other two are multi-cavity, semi-automatic. The weight of each of the hand molds is approximately 30 pounds, whereas the weights of the two semi-automatic molds are 118 and 300 pounds respectively. For the pieces concerned, all of which are molded from wood flour filler material, the application of the flash overflow type of mold is correct, as is the multi-cavity classification for two of the molds mentioned. The use of the semi-automatic classification, however, is not justified by the production required for the particular pieces. It is of interest to note the excessive weight of the semi-automatic, multi-cavity molds as compared with the single cavity hand mold.

Figure II shows two typical designs of molded pieces which are suitable for production using flash overflow molds.

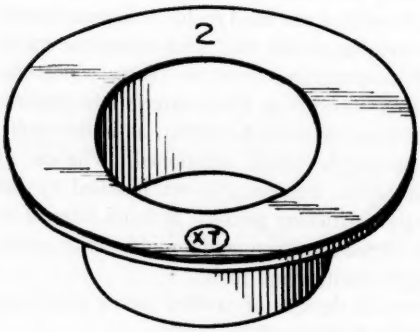
FLASH SUBCAVITY MOLDS.

The subcavity mold as indicated by its name is a form of mold (multi-cavity) which has a common loading chamber for all cavities, the cavities being located in the bottom of the mold. The common loading chamber is of sufficient depth to accommodate the multiple charge. This use of a single bulk charge obviates the necessity of separate weighing of the material for each cavity, as is required for all the other types of molds. In this type of mold the force bottoms directly on the metal land surface, forcing the molding material into the several subcavities.

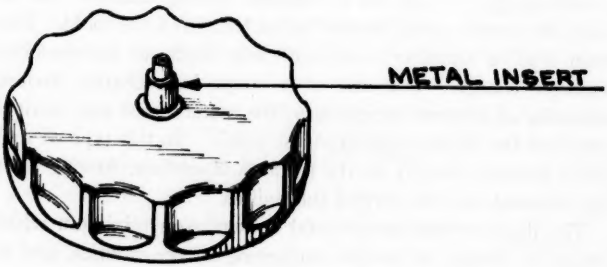
The flash or land area should be kept to a minimum within the limits of design, by proper staggering of the cavities, and should not exceed one-half the cavity area. Subcavity molds may be flash or positive molds, but are generally positive type.

This type of mold should be restricted to the production of thin simple pieces of small size (not over one inch in diameter), when the material to be molded cannot readily be preformed, and when the plasticity of the molding material is very high, the electrical and physical requirements for the finished parts are very moderate, and the factor of cost is of major importance.

The flash subcavity type of mold has no particular application in



MOLDED BUSHING



MOLDED CAP

FIG. 11.

the production of pieces for Naval installations, although there have been molds of this type used.

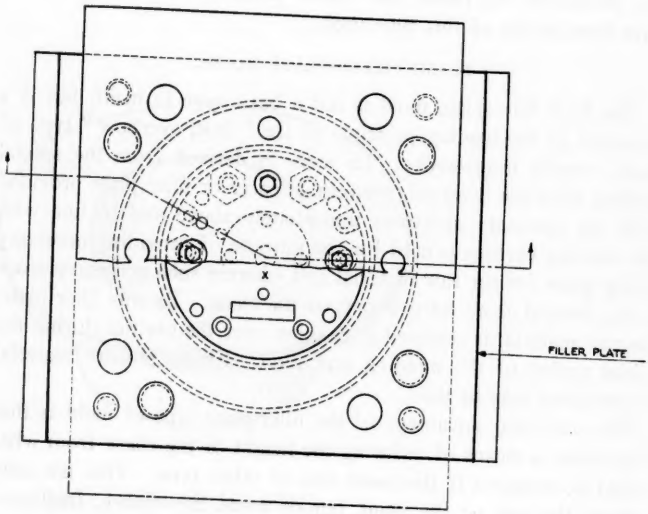
FLASH FILLER-PLATE MOLDS.

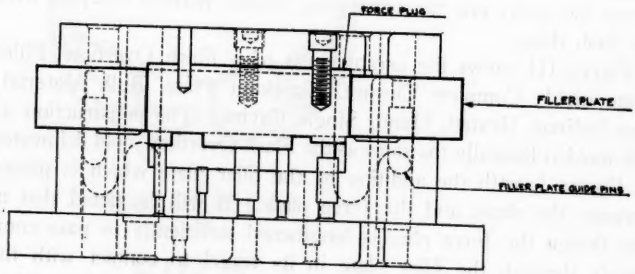
The flash filler-plate mold is not a basic type in itself, but is a variation of the loading principle of the "flash overflow" type of mold, usually multi-cavity. Its name is derived from the special loading principle involved, wherein a spacer or filler plate provided with the necessary apertures immediately above and in line with the molding cavities is used for the purpose of providing necessary filling space for the raw material and wherein such necessary space is not formed as an integral part of the chase. By this filler plate, the raw material is confined in position over the cavities during the initial period of the molding operation. Either powder material or preforms may be used.

The particular advantage of the filler-plate type of mold is that it provides a means of reducing the height of the chase from what would be required if the mold was of other type. This not only reduces the cost of the mold, but of more importance, facilitates the placing of inserts in the cavities, and reduces the probabilities of breakage of the insert holding or hole forming pins.

In the application of the filler plate to the flash mold, the mold is loaded with the filler plate in contact with the flash ridge. As the force plug descends, the filler plate confines the material between the cavity and the force plug, excess material escaping over the flash ridge.

Figure III shows the arrangement of a Flash Overflow, Filler Plate mold; Complete Closure, Knockout Plate, Bulk Material; also Indirect Heated, Hand, Single Cavity. The construction of this mold is basically the same as the Flash Overflow mold delineated in Figure I, with the addition of the filler plate which is placed between the chase and the force plate. It will be noted that in this design the force plug is lengthened sufficiently to pass completely through the filler plate in its travel to contact with the chase. In this mold the force plug is recessed in the force plate ⅛ inch, differing from the flash overflow mold in this respect. The force plug also contacts with the flash ridge, as the force plate is separated from the chase the distance of the filler plate thickness.





UNIT CONSTRUCTION, INDIRECT HEATED, HAND, SINGLE CAVITY, FLASH OVERFLOW, FILLER PLATE, COMPLETE CLOSURE, KNOCKOUT PLATE EJECTING MOLD.
FIG. 111

The filler plate has an individual set of guide pins which holds the plate in proper position, independent of the action of the force plate.

Molds of the flash filler plate classification have been furnished having the forms of "Spacer Fork" and "Spring Box" listed under the "Closure" classification. These particular molds are designed for the use of C. F. I. material which is contrary to a previous statement made in the foregoing that flash molds are not suitable for use with fabric base high impact materials. However, this is just another instance of what may be expected in matters concerned with the molding of phenolic resinoids, where we find that any definite rule of procedure that may be attempted will resolve itself into a rule to be broken in specific cases

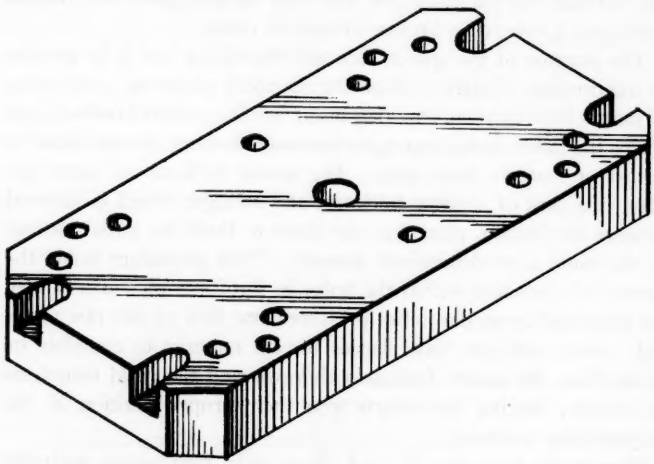
The purpose of the spacer fork and the spring box is to provide an intermediate closure position for the mold where the positioning of inserts held by relatively long insert holding pins is involved, and where the lateral pressures upon the insert holding pins is liable to distort or fracture these pins. The spacer fork, as its name implies, is a fork of suitable thickness and strength which is inserted between the bottom plate and the chase to limit the initial closure of the mold a predetermined amount. This procedure holds the inserts in a position within the holes in the force plug where they are protected from the initial pressures and flow of the raw material. After sufficient time elapses for the material to complete its initial flow, the spacer fork is withdrawn and the mold completes its closure, forcing the inserts into their proper position in the polymerizing material.

The spacer fork may be used alone or in conjunction with the spring box. The spring box provides a means of automatically forcing the chase and bottom plate apart, to provide the necessary opening for removal of the spacer fork, when the pressure is released and the top force is opening the mold. This type of construction relieves the press operator from the necessity of manually parting the mold, thus leaving his hands free to remove the spacer fork.

The arrangement of a mold having spring box and spacer fork features is shown in Figure VII.

For naval production, the flash filler plate mold has had quite a wide application. There are now in active use flash, filler plate

type molds for producing possibly twenty-five Navy standard designs of molded pieces; of this number, seven are single cavity hand type, five are single cavity, semi-automatic type and the remainder are multi-cavity, semi-automatic type. Under the latest ideas concerning the proper application of mold designs, six of the seven hand, flash filler plate applications are questionable and in event of purchase of replacements of these mold equipments in all probability would not be duplicated as such. These six hand mold applications are designed for C. F. I. material and although the molds can be operated, difficulties have been experienced with them. Their chief interest lies in the fact that they represent an effort in the



MOLDED BASE

FIG. IV.

recent transitory stage of mold designing to lead away from the known limitations of the truly positive mold for Navy work. The single cavity, semi-automatic molds of this group are quite heavy (weighing from 200 to 300 pounds each) for handling and shipment by the custodian Navy Yard, and although they can be

operated satisfactorily, it has been demonstrated that sufficient production with greater ease of handling and under better economic conditions is obtainable with the use of possibly two or three 4-cavity hand molds.

Four of the semi-automatic, multi-cavity molds of this group fall in the same category as the single cavity molds referred to, except that the weights of these molds are excessive to the point of being prohibitive. One of these molds weighs 750 pounds and another, 400 pounds, making handling, shipment and inspection an extremely difficult matter. Unless other considerations enter into the matter, attempt should be made to limit the weights of Government-owned molds to a figure—possibly 200 pounds maximum, so that the handling of the mold will not become an economic difficulty.

There are eight semi-automatic, multi-cavity molds in this group which are of a particular construction, developed to meet a certain production for receptacle plug tips. This application, although contrary to some accepted ideas for mold applications, appears satisfactory, and falls in that category where the exception proves the rule.

Figure IV shows the type of piece successfully produced by the flash filler plate type of mold.

FLASH STRIPPER PLATE MOLDS.

The flash stripper plate mold is, as the name implies, a type of mold fitted with a "plate," the function of which is to eject the molded pieces from the mold. A stripper plate mold may be flash, or positive, but is generally of the flash type. The advantage to be gained by the use of this type of mold lies in the rapid ejection of the molded pieces.

This type of mold, in its simplest form consists of parts of the usual flash design as enumerated for the flash overflow mold and in addition, a stripper plate. The force plate and stripper plate operate independent of each other.

During the loading operation, the stripper plate may be retracted in contact with the force plug, or it may be in position over the mold cavity, in which case the mold cavity is charged through the stripper plate, acting as a "filler plate." The force plug, in descend-

ing, initially traps the charge in the mold cavity and thereafter acts as a displacement plunger, forcing the excess material over the flash surface. Upon opening the mold, the plunger and stripper plate ascend as a unit, the molded part being carried out of the chase by the force plug. At the proper point, the motion of the stripper plate is arrested, and the continued motion of the force plug causes the stripper plate to remove the molded part from the force plug.

In some cases stripper plate molds are provided with knockout pins, to permit the clearing of any cavities where the automatic ejecting fails to function properly.

Flash stripper plate molds may be loaded either with preforms or with powdered material, depending upon the design of the piece.

This type of mold is suitable for the commercial molding of relatively simple pieces having thin bottom sections, in multi-cavity applications where the use of ejector pins might deform or puncture the bottom of the pieces.

For the molding of Navy designs of phenolic resinoid pieces, the flash stripper plate mold offers no advantages that cannot be obtained with other forms of molds. Moreover, as far as can be foreseen, designs will not usually be developed having such thin sections that ejection of pieces cannot be accomplished by the use of knockout pins.

There are a few so-called stripper plate, filler plate, flash molds in service which produce Navy bobbin-type switch handles. These molds have apparently produced satisfactory pieces in the past, but there has been no recent production from these molds under the present more rigid test requirements to form a proper basis for discussion as to their merits.

Coming now to the positive molds, it is of particular interest to note that the basic principle of this type of mold is quite different from that of the flash type. As previously mentioned flash molds have their cavities cut in the chase to the exact depth of the finished piece. In positive molds, on the other hand, the cavities are formed deep in the chase, the aperture thus formed above the cavity, constituting the loading chamber and acting as a pressure cylinder in which the force plug descends.

Positive molds may be generally characterized as a type of mold in which the force plug telescopes with the mold cavity.

Under the classification of "positive molds" are found the following subtypes:

Truly positive
Landed positive
Semi-positive
Sub-cavity.

TRULY POSITIVE MOLDS.

The truly positive type of mold is of a type distinct in itself. There being no restriction to the movement of the force plug in the pressure cylinder, the full molding pressure is exerted positively upon the molding charge and upon the molded piece at the final closure of the mold. Hence the name "truly positive." There is a close analogy between a truly positive mold and a piston in a cylinder.

With a given mold pressure, the final dimensions of the molded piece depend upon the weight of the charge. There is required, therefore, an exact measurement of the mold charge to obtain accuracy and uniformity of the dimensions of the finished pieces.

Truly positive molds require a relatively deep chase to provide sufficient loading area within the pressure cylinder above the cavity. In general, this area must be from 2 to 8 times the finished piece depending upon the type of material used. There is only a slight vertical fin formed on the finished piece with this type of mold.

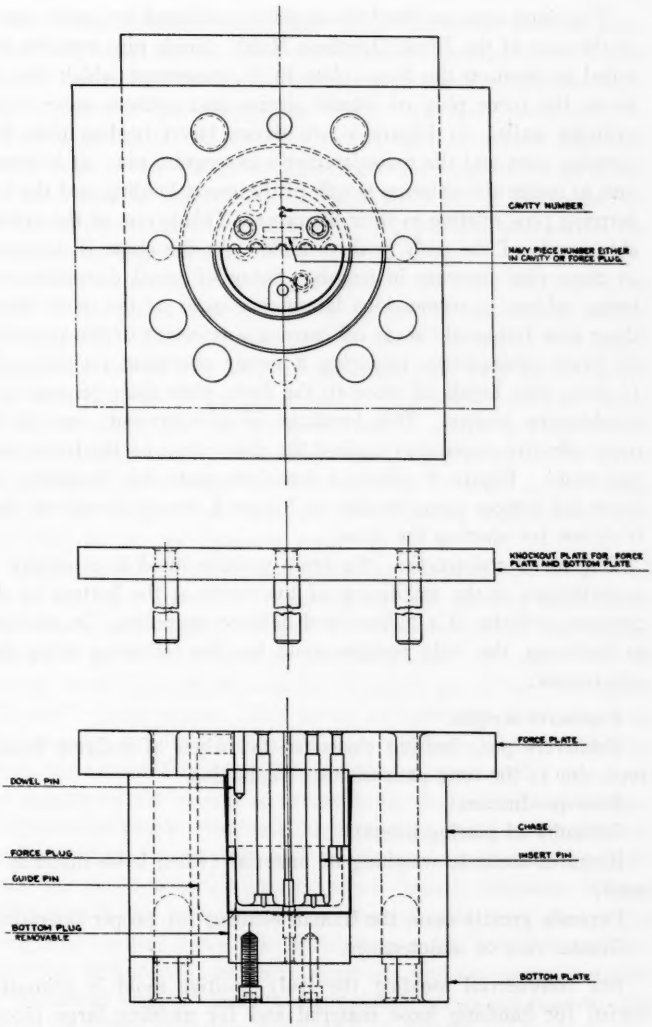
Loading the truly positive type of mold, for the powder materials, is a simple matter, as the exact amount of material is merely poured into the aperture above the cavity. For the fabric base, high impact, materials the loading of the mold introduces a serious design problem due to the high bulk factor of the material. To provide sufficient area in the loading aperture for this high bulk factor requires a very deep chase which adversely influences the heating and operating characteristics of the mold as well as the weight and cost. High impact materials, therefore, used with truly positive molds preferably should be preformed.

Ejection of the piece in the truly positive mold is generally done by means of knockout pins. In some instances the lower force plug may be made removable, and the piece ejected by moving the chase relative to the upper or lower force.

Figure V shows the arrangement and principal design features of the Truly Positive Mold, Built-Up Construction, Indirect Heated, Hand, Single Cavity, Complete Closure, Knockout Plate, as required for the production of molded pieces of high impact material for Naval work. The increase in the overall height of the mold assembly over Figure I is noticeable. The force plate and bottom plate, it will be noted, are similar to the Flash Mold. The force plug is recessed in the force plate ⅛ inch, and is doweled and secured to this plate by machine screws as in the case of the Flash Filler Plate Mold, Figure III.

The force plug telescopes with the aperture in the chase and as will be seen conforms in its shape to the piece to be molded. It has been the practice for this type of mold, in order to ease the movement of the force plug within the chase, to have the cylindrical surface of the force plug relieved by a series of flattened faces spaced equally; a bearing face the full diameter of the force plug is left between the flattened faces of not less than 3/16 inch. These flattened sections do not extend closer to the molding face of the force plug than 3/16 inch and extend to within not less than 3/16 inch of the top of the force plug. As a substitute for this, it has lately been agreed that for round pieces requiring deep draws, the chase of the mold should have a taper of about 1 degree to a point ⅛ inch above the position of the piece when molded; above this point the chase is straight.

It will be noted that the bottom of the cavity is formed by a bottom plug set deeply in the chase, which is secured to the bottom plate in the same manner as the force plug is secured to force plate. The area within the aperture above the bottom of the cavity is shown in the ratio of 5.2 to 1 with that of the finished piece. This ratio or bulk factor has been found satisfactory for C. F. I. loose material so far as loading the mold is concerned. Very recently a decision has been reached that since Type C. F. I. material can be used in special loose preforms, and moreover since the use of these preforms will improve the quality of the piece, this type of mold may be based on the reduced bulk factor of 2.5 to 1. In this case the die for these special preforms will be considered a part of the mold equipment. This decision will have a far-reaching and beneficial effect on the design of the truly positive mold.



BUILT UP CONSTRUCTION, INDIRECT HEATED, HAND, SINGLE CAVITY, TRULY POSITIVE, COMPLETE CLOSURE, KNOCKOUT PLATE EJECTING MOLD.

FIG. V.

The chase rests on the bottom plate, positioned by guide pins as in the case of the Flash Overflow Mold. Guide pins are also provided to position the force plate in its movement which also relieves the force plug of lateral strains and reduces wear in the cylinder walls. In Figure V are shown insert holding pins, hole forming pins and the manufacturer's monogram pin. It is important to notice the extreme length of the insert holding and the hole forming pins relative to their diameter. This is one of the vulnerable points of the truly positive mold. As the chase is deepened, so these pins increase in length. Being of small diameters, and being subject to tremendous lateral pressures as the mold closes, these pins frequently shear off causing a rejection of the piece and of more consequence, requiring a repair operation on the mold. If these pins break off close to the force plate their renewal is a troublesome matter. This breakage of pins presents one of the most effective arguments against the deep chase of the truly positive mold. Figure V shows a knockout plate for removing the force and bottom plates similar to Figure I, but no knockout plate is shown for ejecting the piece.

Due to its construction, the truly positive mold is expensive to manufacture as the machining of the cavity at the bottom of the pressure cylinder is a tedious and delicate operation. In addition to high cost, the truly positive mold has the following other disadvantages:

Excessive weight.

Relatively poor heating characteristic, when of indirect heated type, due to the long path of heat conduction.

Slow production.

Difficulty of placing inserts.

Requires accurate weighing of material (when loose material is used).

Depends greatly upon the human element for proper operation.

Greater cost of maintenance.

For commercial molding the truly positive mold is primarily useful for handling loose material and for molding large pieces containing numerous inserts, where it is impracticable to produce preforms and where the loading time is a small fraction of the entire molding cycle.

For Naval production, in spite of all the disadvantages mentioned, the truly positive mold stands as the one type of mold that has the ability to produce molded pieces of the high uniform density required to pass inspection tests. Its chief weakness for Naval work lies in the possibility, when loose materials are used, of variation in the weight of the charge "creeping in," even under the most careful supervision. The enforced use of preforms for all Naval production is one answer to this problem and merits the most careful consideration.

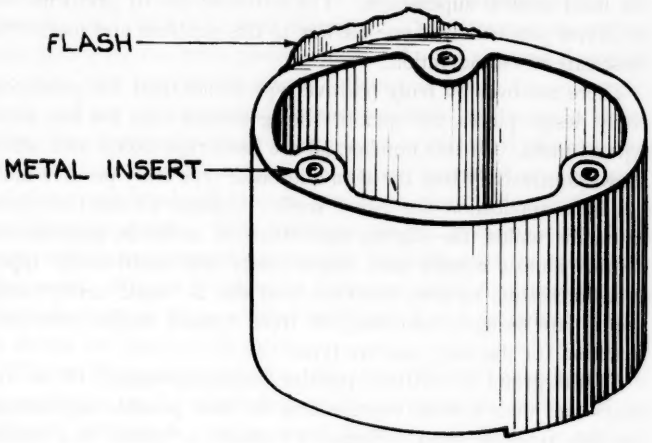
There are over 50 truly positive type molds used for producing Navy design pieces, this quantity being greater than for any other type of mold. Of this number, all are hand type except two which shows conclusively that the semi-automatic type truly positive mold has little application for Navy work. Outside of the two semi-automatic molds, the slightly more than 50 molds in question are divided almost equally into single cavity and multi-cavity types. It is interesting to note, however, that the 25 single cavity molds are all late designs, indicating the trend toward single-cavity construction for the truly positive type.

Of the group of 25 truly positive hand-type molds, 18 are for boxes and their covers, representing the best possible applications for this type of mold. Figure VI shows a typical box design produced by one of these truly positive molds. The molds for all except three of the boxes and for all covers are of the "Complete Closure" classification. This group of molds are for what may be termed "smaller" boxes and covers whose largest dimension does not exceed 6 inches. The weights of these molds run from 55 pounds to 200 pounds according to the size of the molded box.

The three molds mentioned as exceptions are for larger boxes and are of the spacer fork and spring box classification to facilitate the positioning of inserts which would otherwise involve difficulties if the complete closure type of mold were used. The weight of each of these three spring box molds is in the neighborhood of 300 pounds, and here again we find an exception to the "hand mold weight" limitation, previously set at 100 pounds. No other type of mold, however, would be suitable for these designs.

Referring again to Figure V, showing a truly positive mold for a box such as Figure VI, it will be noted that the box is in a posi-

tion with the bottom down. It will be seen that if the box were molded in a position with the bottom up, the holding pins for the inserts located in the bottom of the box would be much shorter; a very desirable feature. If the box were in this reversed position the deep side wall of the box would be formed in the chase instead of being formed in the force plug, and the depth of the chase



MOLDED BOX

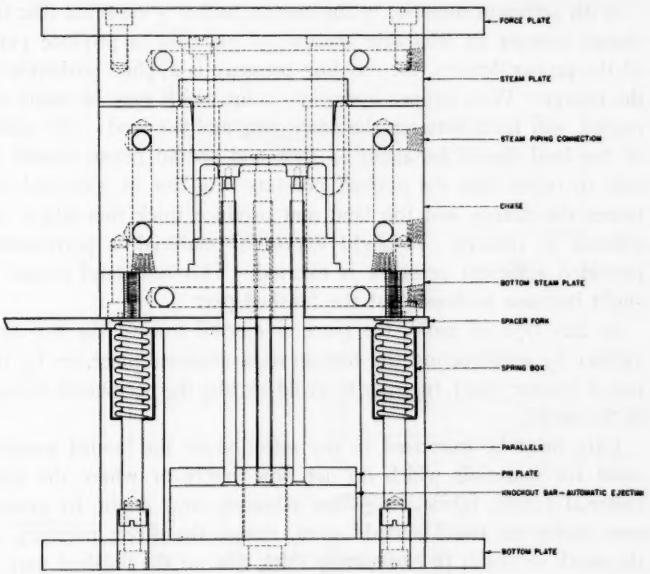
FIG. VI.

would be increased to provide the necessary loading area. For some box designs and using some high impact materials the molding of boxes, bottom up, has proven very satisfactory and such procedure is highly recommended by some molders. On the other hand, much opposition has developed over this method and as all molders *can* mold boxes, bottom down, the bottom down position has been adopted for molding boxes in truly positive molds, as shown in Figure V. This point is mentioned in view of the highly controversial nature of the problem.

The nine cover molds of this group show more uniformity than the box molds, all being of the complete closure classification.

Their weights run from 40 to 165 pounds, generally in accordance with the size of the molded piece. The remaining quantity of truly positive molds of this group are for producing insulating bases and follow closely the arrangement of the cover molds.

The group of approximately 25 multi-cavity truly positive molds previously mentioned as coming under the category of older designs, includes a number of mold types, for producing quite a varied number of molded designs. In this group we have switch handles, switch spacers, lamp sockets, insulating bases, push-bottom bodies, spools and even one case of box and cover. Some of these applications, notably switch spacers and lamp sockets, are correct in their multi-cavity classification even if of truly positive type. A number of these molds, however, if replaced would not retain the multi-cavity classification. Where the use of multi-cavity molds is



BUILT UP CONSTRUCTION, DIRECT HEATED, SEMI-AUTOMATIC, SINGLE CAVITY, TRULY POSITIVE, SPRING BOX AND SPACER FORK, AUTOMATIC EJECTING MOLD.

FIG. VII.

desirable, four-cavity seems to fulfill the present ideas as to the extent cavities are to be "multiplied" for Navy work, whereas, a few years ago molds of 12-cavity and even 24-cavity were advocated.

The arrangement of a truly positive mold, semi-automatic, single cavity, direct heated, spacer fork and spring box, automatic ejecting is shown in Figure VII.

LANDED POSITIVE MOLDS.

The landed positive mold differs from the truly positive type in that there is a "land" or shoulder in the compression space just above the molding cavity. This type of mold insures a uniform height of the molded piece by reason of this land. The landed type of mold is somewhat less positive in its action than the truly positive type, although the principle of trapping the material is retained.

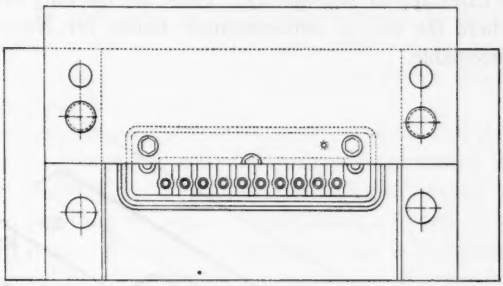
With sufficient material in the charge, and it is essential that the charge contain an adequate amount of material to produce parts of the proper density, the molding pressure is applied positively to the charge. With proper operation, a fin which may be easily removed, will form between the force plug and the land. The width of the land should be about $\frac{1}{8}$ inch and should never exceed $\frac{3}{8}$ inch in order that the applied pressure will not be absorbed between the charge and the land and produce thick fins which are difficult to remove. A slight excess of material is permissible, provided sufficient pressure is exerted. This will tend toward a slight increase in density of the finished part.

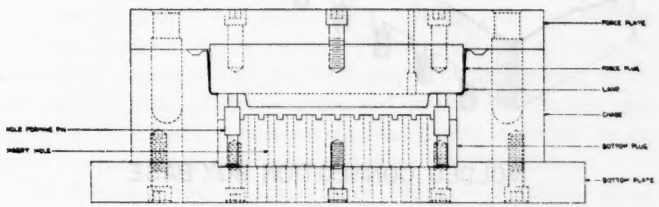
In this type of mold, the piece is ejected toward the top face (either by constructing the bottom as a separate piece, or by the use of ejector pins), in order to avoid scoring the peripheral surface of the mold.

Care must be exercised in the selection of the landed position mold for materials which do not flow freely or where the filler material (cloth, fabric, long-fibre asbestos, etc.) might, by premature curing on the "landed" area, reduce the direct pressure on the work or result in excessively thick fins on the finished part at the dividing line of the mold. The selection between the truly positive or landed positive type of mold also depends upon the most desirable position of the flash—vertical or horizontal—and upon the resultant cleaning operation. The shape of the piece and

the position in which the piece is molded will enter into the decision as to the preferred type. Usually the manufacturer will be requested to submit his reasons for the use of the landed positive type of mold, when proposed, for the particular item that may be in question.

Figure VIII shows the design and arrangement of a Landed Positive Mold; Built-Up Construction, Indirect Heated, Hand, Single Cavity, Complete Closure, Knockout Plate Ejecting. The construction of this type of mold, except for the "land" is much the same as the truly positive type and needs no detail description. The location of the "land" is well ilustrated by this sketch, and it will be seen that the flash found in the piece will be horizontal in its direction.

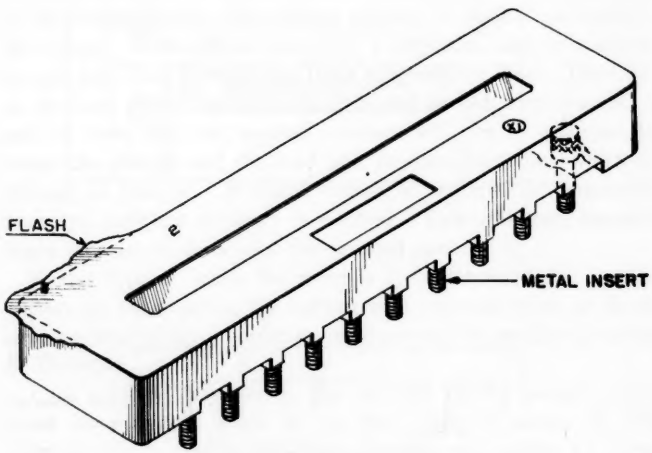




BUILT UP CONSTRUCTION, INDIRECT HEATED, HAND, SINGLE CAVITY, LANDED POSITIVE, COMPLETE CLOSURE KNOCKOUT PLATE EJECTING MOLD
FIG. VIII

Next to the truly positive type, the landed positive mold has the most numerous applications in the production of Navy design molded pieces. There are approximately 25 single cavity molds and 20 multiple cavity molds of this type in active use. Of the group of single cavity molds, all but one are of the hand classification. The one semi-automatic mold mentioned will undoubtedly revert to hand type in event of a replacement mold being purchased.

The group of landed positive multiple cavity molds has quite a considerable variation of cavity arrangements, running from 2-cavity to 12-cavity. The 4-cavity arrangement predominates and is believed to be the most satisfactory set-up for Navy molds. Seven molds of this group are semi-automatic, and the applications in these cases appear satisfactory. These are the only cases, however, where the use of semi-automatic molds for Navy work is found justifiable.



MOLDED CONNECTION BOX BASE

FIG. IX.

Weights of the landed positive type of molds are comparable with the weights of similar truly positive molds.

Figure IX shows a design of molded piece found suitable for production from a landed positive type of mold.

SEMI-POSITIVE MOLDS.

The semi-positive mold is a composite type, combining the features of the positive and flash molds. This type of mold is even less positive in its action than the truly or landed types. It operates as a flash mold within a short distance of final closure, when, at this point, the force plug telescopes within the chase just a sufficient distance to exert a positive pressure on the charge during the final closure of the mold. Compared to the positive and landed positive types, the total mold height is much less, due to the short force plug. In some types of semi-positive molds, in addition to the flash surface at the top, a slight clearance between the force plug and the lower die is provided which permits excess material to escape, forming a thin vertical fin.

If no clearance is provided between the lower die and the force plug, the mold operates as a landed positive mold during the last fraction of an inch (1/32 inch to 1/8 inch) of travel. When clearance is provided, by proper proportioning of parts, the thickness of the vertical fin may be determined, and is independent of the thickness of the flash, which is advantageous in multiple-cavity molds. Extreme care must be exercised in this type of mold that the clearance between the lower die and the plunger is not too great, otherwise the mold becomes a flash mold, the advantages of a semi-positive mold are lost and the extra construction is wasted.

The semi-positive type may also be of the stripper-plate type. This type of mold is well adapted for molding from tabletted or preformed material and using multiple-cavity molds, and may be successfully employed with the more plastic of the molding materials and for parts of simple shape and small size. Its use may provide good economy under some circumstances, provided its inherent limitations are understood and given due consideration.

The Navy has four applications of semi-positive molds, all of which molds are designed for the production of boxes of high impact material. These molds are single cavity hand type and are

fitted with filler plates and also use spacer forks. The force plug of these molds telescopes with the cavity only a distance of 1/16 inch, which represents the distance that the feet of the boxes molded extend below the bottom surface of the box.

These molds represent a particular development for the group of boxes concerned, where the boxes are molded "bottom up" and where a particular type of high impact material was used. The reason back of the development was sound, which was to avoid the use of long insert holding pins with high impact material. From a weight and cost standpoint the molds were an attractive proposition and thus operation under the conditions designed for was satisfactory. Due to differences of opinion arising among the manufacturers regarding these molds, it may be that further molds of this type will not be purchased.

POSITIVE SUB-CAVITY MOLDS.

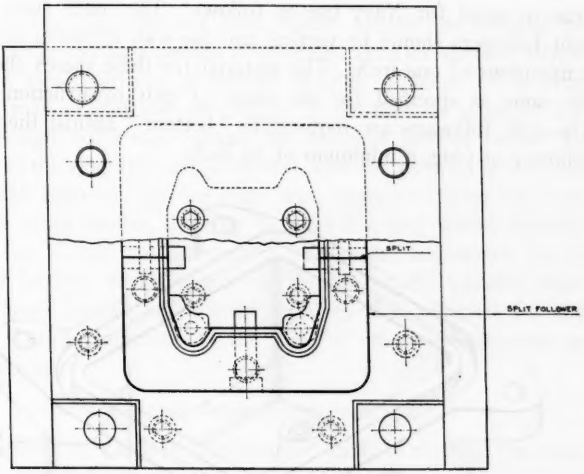
Positive sub-cavity molds were mentioned under the "Flash Mold" classification and the description under that heading is sufficiently complete to form an idea as to the operation of this type. As the likelihood of this type of mold being used for new production is small, no further discussion of this type is considered necessary.

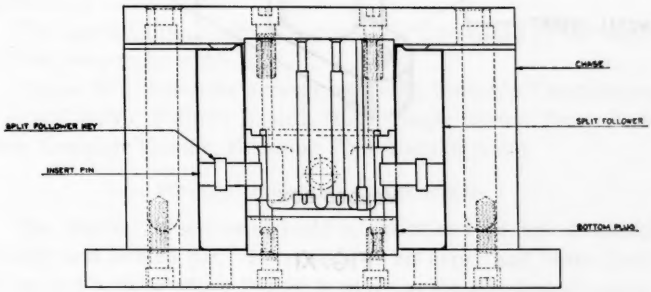
Under the "Construction" classification there are listed three types of molds not yet touched upon. These types are the Split Follower, the Ring Follower, and the Floating Chase which are found as modifications of the positive types of molds.

POSITIVE SPLIT FOLLOWER MOLDS.

The split follower molds is usually landed positive. This type of mold makes possible the molding of pieces having projections of such form that would prohibit the removal of the finished piece from a "solid" chase mold.

The cavity bearing number or "follower" is split into two or more pieces and when assembled, fit tightly into an aperture cut in the chase to receive it. The follower sections are keyed together. The mold is operated the same as the basic type except that to remove the molded piece, the entire follower is ejected as a single unit, then opened for the ejection of the piece.

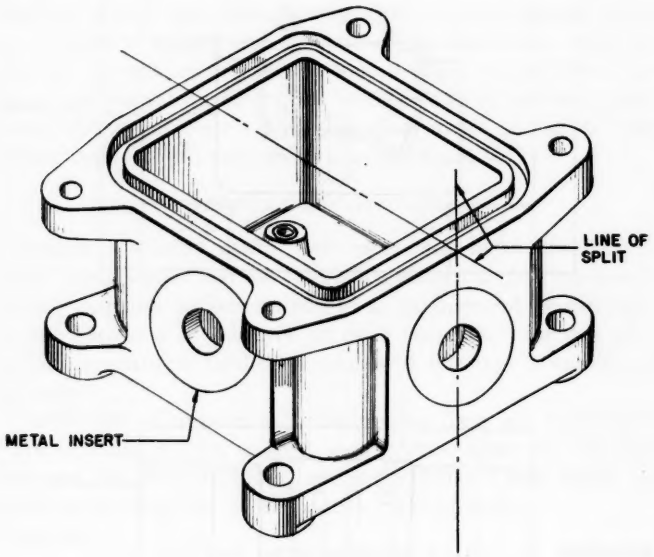




BUILT UP CONSTRUCTION (SPLIT FOLLOWER) INDIRECT HEATED, HAND, SINGLE CAVITY, TRULY POSITIVE, COMPLETE CLOSURE, KNOCKOUT PLATE EJECTING MOLD.

FIG. X

Certain requirements have been developed for the split follower type of mold for Navy use as follows: The outer sides of the split followers should be vertical and the wall thickness should be a minimum of one inch. The material for these pieces should be the same as specified for the chase of unit construction molds. The split followers are required to "bottom" against the bottom follower or plug, a minimum of 5/8 inch.



MOLDED BOX.

FIG. XI

Figure X shows the arrangement of a Built-Up Construction (Split Follower), Indirect Heated, Hand, Single Cavity, Truly Positive, Complete Closure, Knockout Plate Ejecting Mold. Figure XI shows a typical box with projecting lugs requiring the use of the split follower type of mold.

POSITIVE RING FOLLOWER MOLDS.

The positive ring follower mold makes possible the molding of circular pieces having a section of threaded exterior surface, which otherwise would involve a difficult ejection problem. The ring follower may extend just the length of the threads as called for by the design, or it may enclose the complete molded piece. In either case the ring follower is ejected with the piece as is the case of the split follower, and the piece then unscrewed from the ring.

As the name implies, the ring follower is a ring whose exterior surface fits closely the aperture in the chase, and which has a threaded interior which forms the threads of the molded piece. The follower should be of steel equal to that required for the cavity. The performance of the mold is otherwise the same as the basic type.

POSITIVE SPLIT-CAVITY MOLDS.

In certain instances it is found impracticable to machine the chase or die to the shape of the piece to be molded, and yet no difficulty in ejecting the piece is involved that would require the use of the split follower type of mold. For such applications a built-up construction mold is used with the die made in two or more pieces. These separate pieces are then forced into the chase, forming a permanent construction.

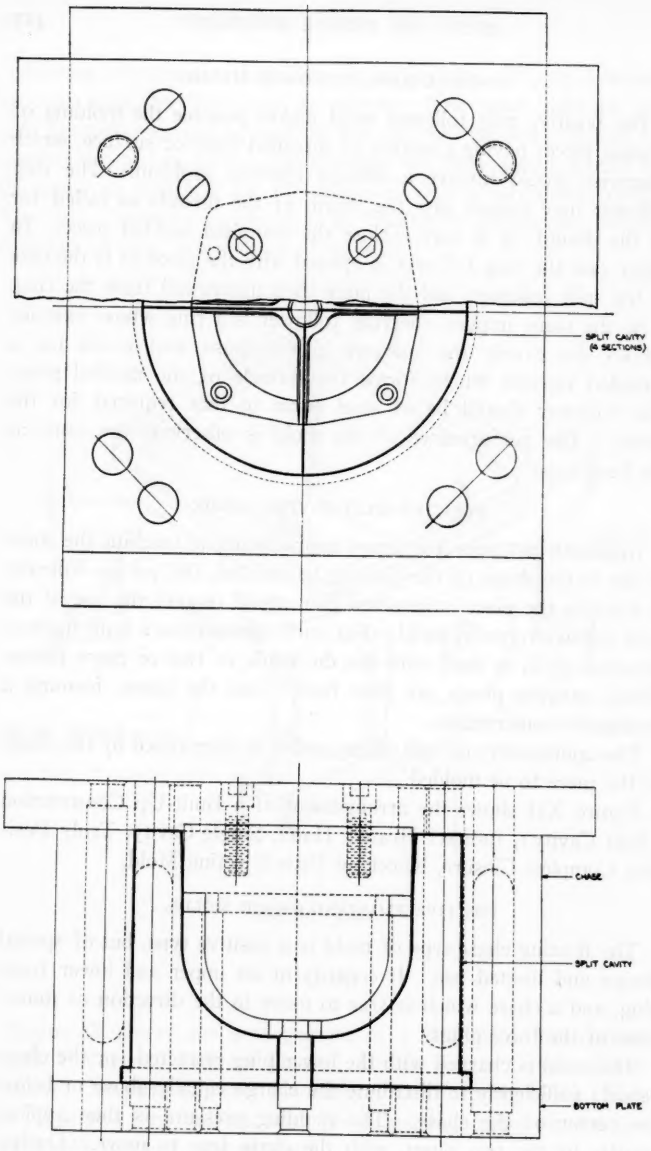
The applicability of split-cavity molds is determined by the shape of the piece to be molded.

Figure XII shows the arrangement of a Built-Up Construction (Split Cavity), Indirect Heated, Hand, Single Cavity, Truly Positive, Complete Closure, Knockout Plate Ejecting Mold.

POSITIVE FLOATING CHASE MOLDS.

The floating chase type of mold is a positive type, but of special design and limited use. It consists of an upper and lower force plug, and a chase which is free to move in the direction of movement of the force plugs.

The mold is charged with the lower plug retracted (or the chase raised) sufficiently to distribute the charge equally above or below the center of the chase. The molding pressure is then applied equally by the two plugs, with the chase free to move. During



BUILT UP CONSTRUCTION (SPLIT CAVITY) INDIRECT HEATED, HAND, SINGLE CAVITY, TRULY POSITIVE, COMPLETE CLOSURE, KNOCKOUT PLATE EJECTING MOLD

FIG XII

the molding operation the chase floats to a position of equilibrium with respect to the friction of the molding charge, so that the same molding pressure is applied at each end of the piece, and variations in density of the molded part are materially reduced.

This type of mold is primarily applicable to the molding of pieces whose vertical height is great with respect to the horizontal dimensions, when such height is sufficiently great to cause appreciable variation in density from top to bottom, if molded in an ordinary positive mold. It is primarily intended for use with powdered material, but in special cases may be applicable to the molding of tabletted material. This type of mold may have its application for Navy work but may be considered, at least for the present, as having a very limited application.

EXTRUSION (TRANSFER) MOLDS.

The extrusion type mold, also referred to as the "transfer molding method," consists essentially of a cavity into which the molding composition is forced while in a plastic condition, by the "extrusion" or "transfer pressure" method. The composition is forced through one or more small orifices from a pressure chamber into the mold cavity.

Special machines are required for extrusion molding of thermoplastic materials. A loading chamber with automatic feeder is usually provided. Dies are brought into position with channels leading from the loading chamber and filled.

Thermo-setting plastics to date require that a slight charge be placed in the loading chamber each cycle; so, in general, special assemblies and regular presses are used when thermo-setting plastics are molded by the extrusion process.

The extrusion (transfer) type mold is used commercially to a limited extent for molded pieces which cannot ordinarily be molded by other processes, and which pieces have thin wall sections or of extreme irregularity in shape; also pieces having excessively long inserts which might become bent during molding operation.

Study is now being made of this type of molding with a view to determining if there are any advantages in its adoption for the production of Naval designs. It is believed that the usefulness of

this method, should it prove successful, would be in the field of high impact material molding by reason of the increased possibilities of obtaining a more uniform density of the piece than is possible with pressure molding; particularly this would be true for pieces having small projections where pressure molding tends toward a separation of the fabric from the resin.

The uses and limitation of this type of molding will have to be determined from a Naval viewpoint by trial manufacture and subsequent tests of specific designs.

SPECIAL MOLDS.

Combinations and modifications of the fundamental types of molds are possible and are found in the molding industry. Such molds are characterized as "Special Molds." An occasion may arise when for a particular application on a special molding problem such a mold may be required.

BLOWING MOLDS.

Blowing molds consist of simple dies, loaded with sheet or tube stock, which form a chamber of plastic, non-porous material which is expanded against the walls of the mold cavity by internal pressure. Pressure is introduced by compressed air, steam, or the introduction of volatile salts within the plastic chamber, which generate gas pressure when heated.

Blowing molds are used exclusively for the production of hollow molded parts and only for the acetates and cellulose nitrates. Athough the molds have no applications which are now foreseen for Navy work, they are mentioned in order to form a complete category of known mold constructions.

There are certain features pertaining to the design of molds which are generally applicable to all types and classifications and may be considered under the heading of general design characteristics. These items embrace the following:

Ruggedness
Bulk factor
Shrinkage
Draft
Materials.

Due to the enormous pressures under which molds operate, a primary consideration of mold design is ruggedness. Molds therefore, should be very substantially constructed and all parts thereof should be designed with ample strength to resist molding pressure without distortion, and to minimize damage from handling. Too much stress cannot be placed on this requirement, as many cases have been found of Government-owned molds where these molds have distorted or broken while being operated.

The basic molding compounds are much greater in bulk than the resultant molded piece, as previously mentioned. This relation between the bulk of the molding compound and that of the molded piece, which varies greatly with the type of compound, is termed the "bulk factor." The term "bulk factor" is also applied to that characteristic of the mold which concerns the relation of the area of the loading chamber with that of the cavity.

In view of the fact that the bulk factor of the molding compounds varies from a 2-to-1 ratio to as high as an 8-to-1 ratio, it will be seen that the bulk factor of the molding compound to be used has a very important bearing upon the mold design and its cost, as each mold must be designed, in this respect, for the compound for which it is to be used. Procedures are in use governing the loading of molds and designs incidental thereto, that are as varied as the number of mold design in operation and the number of compounds in use. It would appear that the most simple solution of this matter would be for the mold designer to take the area of the finished piece, which is that of the cavity, and multiply this area by the bulk factor of the material to be used, then allowing sufficient loading area, above the cavity, to accommodate the calculated amount of bulk material. It is, however, not that simple, as questions of economy in mold construction, heat transmission, placing of inserts, weight of mold, etc., enter into the problem, which dictates restriction in the metal mass of the mold.

The principal method used to permit reduction of the loading area is to partially compress or "preform" the bulk material into a volume much less than the basic powdered material and of such form as to facilitate easy loading. This preforming or tabletting is extensively used, and the only objection to its use, so far as Navy

designs are concerned, lies in the fact that some sort of auxiliary means are required to preform this tabletting operation, apart from the actual mold.

For C. F. I. bulk material, it has been found that the size of the filling space for molds, producing molded piece not exceeding 9 inches for both length and width, should be a minimum of 5.2 times the ultimate molded piece. Although C. F. I. has a bulk factor of 8 to 1, it may be readily reduced to 5.2 to 1 by a hand pressure or with slight tamping. If the molded piece produced by the mold is greater in both length and width than 9 inches, it is possible to reduce the filling space to less than the 5.2-to-1 ratio, as C. F. I. material may be "heaped up" in so large an area, and compressed somewhat more readily by hand.

For C. F. G. material the filling space should be 2½ times the ultimate molded piece.

For M. F. G. material, the filling space should be 3½ times the ultimate molded piece. For M. F. G. material, for molds for receptacle plug tips, the filling space may be less than this ratio, in which case, a loose preform must be used.

If preforming or tabletting is used, the result is, of course, a smaller, cheaper mold which when once built precludes the use of bulk material and enforces the continued use of tablets. Where such a mold is approved, the drawing of the mold should show specifically the nature of the tablet used.

The design of molds should take into account the mold shrinkage of the material to be employed, in order to produce molded parts to the proper dimensions, within the limits of the tolerances indicated. Where the tolerances required are very close, the question of uniform shrinkage becomes of extreme importance. Samples taken from each cavity of the mold should be carefully checked for dimensions and fit.

Sufficient side-wall draft should be provided in the design of molds, to permit proper opening of the mold, withdrawal of mold pins, and removal of pieces from the mold. Draft is ordinarily from 1 to 3 degrees. For Navy designs it has been decided that for pieces requiring a deep draw, the chase should have a taper of about 1 degree to a point ⅛ inch above the position of the piece when molded, above which it should be straight.

All parts of the mold should be steel of the highest quality and of the most suitable grade for the particular purpose. Specific applications are discussed under the detail discussion of the various parts of the mold.

Coming to the subject of " detail design " of molds, it is well to start with a categorical list of the parts forming the mold assembly, although mention has already been made of many of these parts in discussing mold classifications.

In order to discuss these parts intelligently it is essential that some standard nomenclature be set up for referring to the various items forming the mold assembly. At present there exists such a variation of names applied to mold parts, by the manufacturers, that it has been found exceedingly difficult to refer accurately to these parts as they are now delineated and listed on mold drawings: To illustrate, taking, for example, the flash mold classification, it is found that there are over 50 names in use applying to the component parts of these molds. For instance, the plates of the mold are listed under all of the following designations: top, force, bottom, loading, inner, steam, pin, retainer, filler top pin, bottom pin, end, side knockout, and stripper. Likewise the mold pins have a multiplicity of names, such as: guide, monogram, identification, dowel, location, insert, hole, cavity, mold, impression, locking, support, supporting, strip, stripping, dummy, center, and knockout. In the same way the chase is termed the die, matrix, block shoe, etc.

In general, all molds have the following parts, as will be seen from Figures I, III, V:

Cavity
Chase
Force Plate (top)
Bottom Plate
Force Plug
Guide Pins (top and bottom)
Monogram Pin.

In addition, some molds in accordance with their classification, will have parts as follows; not all molds, however, having all of these parts:

Cavity Plug (Die)
Bottom Plug
Split Followers
Filler Plate
Stripper Plate
Spacer Fork
Spring Box
Hole Forming Pins
Insert Holding Pins
Knockout Pins
Knockout Plate.
Ejection Bar
Steam Plate
Retainer Plate
Pin Plate
Dowel Pin.

In practically all cases, nomenclature of parts, as listed above, will be found applicable to molds as used for Naval production. All of these names selected are indicative of the nature of the piece.

For discussion of the various parts forming the mold assembly, we will start with the cavity wherein the molded piece is formed.

CAVITY.

The cavity is that part of the mold in which the molded piece is formed, and must accurately reproduce pieces within the tolerances specified for the design. Surfaces of the cavity in contact with the charge should be highly polished and properly hardened.

For molds producing Navy designed pieces, whether or not these molds are Government-owned, each mold cavity is required to have the following marking so disposed as to show properly on the finished piece:

(*a*) Navy drawing number or piece number of the piece in size of lettering as required by the drawing.

(*b*) Molder's trade-mark or symbol, contained within a 5/32, 7/32, or 5/16 inch circle; this marking should be carried on a removable plug of a minimum length of not less than three times the diameter.

(*c*) Serial number of the cavity.

CHASE.

The chase, as previously explained, is the main or central body of the mold, which contains the molding cavity or cavities. The selection of the proper steel for the chase of unit construction molds and the cavity plugs, followers, etc., of built-up construction molds, of necessity, should be given the most careful consideration in order that the cavities will produce pieces of extremely close tolerances. No appreciable wear or change in shape under the effect of heat or pressure can be permitted. Opinion differs as to the best type of steel to be used for the application. Some mold manufacturers advocate the use of a high grade, high carbon, oil hardening tool steel, while others believe that equally good results are obtained with a mild pack-hardened, low carbon, machining steel. The comparative advantages and disadvantages of these classes of steels, disregarding specific instances to the contrary, are as follows:

	Advantages.	*Disadvantages.*
Tool Steel.....	Minimum distortion in heat treating.	Difficult to machine, not suitable for hobbing.
	Minimum distortion under molding pressure.	High cost.
	Long life.	

	Advantages.	*Disadvantages.*
Machine Steel.....	Easy to machine.	Must be pack hardened.
	Readily hobbed.	Liable to distort under heat treatment.
	Low cost.	Hard surface over soft core with tendency to crack under use.

After a study of the foregoing, there is little doubt as to the superiority of tool steel, for general usage for manufacture of the chase and die and, unless hobbing is absolutely essential, tool steel should be required, hardened to Rockwell 65 or Scleroscope 75-80.

The forming of the cavity within the chase of unit constructor molds by hobbing is done extensively and, of course, is advocated by manufacturers as an economical measure. It is believed that the application of this method of manufacture should be carefully

scrutinized before approval as the hobbing method entails the use of a special hobbing steel which is not equal to tool steel as applied to the cavity parts of the mold.

For built-up construction molds wherein the cavities are found by the use of cavity plugs, followers, etc., the chase should be of the best quality of machine steel, pack hardened.

As a result of the experiences encountered in the purchase of molds for Navy Standard pieces, and their subsequent operation, it has been found to be highly desirable to require some uniformity in mold construction for the same type of molds. For the chase of single cavity molds certain limiting features are being established which will prove of value in stabilizing designs. These are as follows:

The chase should be of rectangular shape.

For molds where the molded piece is greater than 1 inch (length or diameter) and does not exceed 5 inches, the minimum thickness of wall of the chase will be 1 inch.

For pieces whose length or width is above 5 inches, to a maximum of 8 inches, the minimum thickness of the wall of chase should be 1½ inches; and above 8 inches, the wall thickness should be 2 inches. For molded piece below 1 inch the thickness of the wall of chase should be as approved for the particular case.

The minimum wall thickness in way of guide pins should be ½ inch.

The thickness of metal between cavities for multiple cavity molds should be a minimum of ¾ inch.

Holes bored in the chase to accommodate guide pins and knockout pins should be carefully positioned so as not to weaken the chase against lateral strains during the molding operation.

FORCE AND BOTTOM PLATES.

The top and bottom plates of the mold which transmit the force (pressure) from the press platens to the molding charge via the force plug are termed " force plates " or under a more closer definition, the upper plate may be designated as the " force plate " and the lower plate as the " bottom plate." Force and bottom plates should be sufficiently rigid to prevent distortion during the molding operation.

For Government-owned molds experience has dictated certain desirable requirements for the force and bottom plates, as follows:

The thickness of these plates in no case should be less than ⅞ inch.

The force and bottom plates should be of the same overall dimensions, and the overhang of the plates should be not less than 1 inch.

Projections of the force and bottom plates should have no holes or slots, these plates being secured to the press by clamping.

These plates shall be made of the best quality machine steel, pack-hardened and ground perfectly smooth.

FORCE AND BOTTOM PLUGS.

The plunger or piston which applies the actual force to the molding charge is called the force plug. The force plug forms the upper face of the molding cavity in the completed closure position of the mold.

The force plug should be recessed in the force plate to a depth of not less than ⅛ inch and secured by Allen head machine screws in a manner insuring accurate alignment at all times. Where practicable, and where space will permit, the size of machine screws should be ½ inch diameter. At least 3 machine screws should be used for securing the force plug to the force plate.

In some cases where it is found impracticable to secure the force plug to the force plate by recessing, a method is used involving the use of a center plug, forming a part of the force plug and passing through the full thickness of the force plate. The diameter of this center plug should be not less than 25 per cent of the area of this force plug, and not greater than 1½ inches diameter. The force plug with this arrangement is secured to the force plate by machine screws as described above. The first method described is considered the preferable method and is required for Government-owned molds wherever practicable.

The force plug should be of the same type of steel and of the same degree of hardness as described for the mold chase.

Where a bottom plug is used in the mold construction, the plug may or may not be secured to the bottom plate. When secured, it should be secured in the manner described for the force plug.

The minimum thickness of this plug should be ¾ inch. The bottom plug should be of the same material as the force plug.

Guide Pins—Two sets of guide pins, one top set and one bottom set, are usually provided. The bottom set of guide pins, which are secured to the bottom plate, position the chase as it rests on the bottom plate. The top set of guide pins maintain proper alignment of the top force (force plate and plug) with the chase as the mold closes during the molding operation. A minimum of three pins, top and bottom, are required, with the pins located so that the plates will assemble in one position only, and prevent their interchangeability.

Guide pins in the force plate should be of length sufficient to extend beyond the force plug so as to prevent undue wearing between the force plug and the chase due to improper registration.

The holes in the chase that receive the guide pins should extend all the way through the chase and should have a close fit with the pins. Directly in line with the holes in the chase, and in the opposing plates from the top and bottom sets of guide pins, holes are drilled of the same diameter so as to preclude the possibility of the guide pins bottoming on any chance material falling into these holes.

Guide pins shall be of tool steel or drill rod, oil hardened to Rockwell 65 or Scleroscope 75-80.

The mold manufacturers differ in the methods used securing guide pins to the plates. One method is to "shoulder" the pins and recess them to a depth of from ¼ inch to ½ inch in the plates, against the shoulder, and then to flush rivet. Another method advocated is to use a drive fit through the plates with a slight taper in that section of the guide pins being driven into the plates. A third method is to shoulder the pins and slightly taper, and then force the pins into holes in the plates, the holes not being drilled completely through the plates. It is believed the first method is slightly preferable and this method is being specified for Government purchases of molds.

Guide pins should be ¾ inch diameter for multiple cavity molds and for single cavity molds producing a piece having a length or diameter exceeding 2 inches. Guide pins for other molds should be of size as approved for the specific case.

Mold Pins—The mold should be provided with the necessary number of mold pins for holding inserts and forming holes as required by the design of the molded piece. These pins are of various sizes, lengths, and shapes, and in general may be classified, as follows, according to the duty performed:

Insert holding pins (parallel to direction of travel of force plug).
Hole forming pins (parallel to direction of travel of force plug).
Knockout hole pins.
Insert holding pins (90 degrees from direction of travel of force plug).
Hole forming pins (90 degrees from direction of travel of force plug).

Insert holding pins should be rigidly secured in the molds and the mechanical assembly of these pins should be such that the pressure exerted on the pins will not be directly applied to the platens of the press, but will be taken up first by the chase, the force, or the bottom plate.

Mold pins should be formed of the best quality hardened tool steel or drill rod, oil hardened. Surfaces in contact with the charge should be highly polished. In general, pins should be so hardened slightly less than parts of the mold that they contact.

INSERT HOLDING PINS (PARALLEL TO DIRECTION OF TRAVEL OF FORCE PLUG).

The diameter of insert holding pins should be larger than the outside diameter of the inserts except where it is required that the molded material make definite flush contact with the exposed end of the insert. The position of the molded piece should always be such that insert holding pins will be of minimum length compatible with other design considerations.

Due to the breakage of insert holding pins, the occurrence of which may be frequent during molding operations, these pins should be so assembled in the mold that they may be readily removed and replaced without requiring a major repair of the mold and a consequent interruption in the molding production.

Figures I, III and V show these pins shouldered and riveted in place which, from experience to date, appears to be the most

generally acceptable method. If in riveting these pins, the ends of the pins are only "headed" enough to hold them rigid, they can be readily knocked out in event of breakage, which arrangement should satisfy the required replacement feature.

Other methods of securing these pins are advocated by the manufacturers, and it is recognized that due to their varied application, one specific method of securing these pins cannot be made universal. Possibly the most satisfactory substitute method of securing these pins, particularly when they are mounted in the cavities or in the force plug, is the "shoulder type drive" fit which involves the use of a shouldered pin with the "shoulder end" having a slight taper. A through hole of smaller diameter is provided in the cavity or plug carrying these pins, through which a drive pin can be inserted to force out the pin if broken. One other method in use where pins are over ⅜ inch diameter, is to embed them in the plates a distance equal to the pin diameter and then secure by machine screw. The hole for the machine screw serves as a knockout hole in the event of the pin becoming broken.

According to the location of the inserts required for the molded piece, insert pins are located in the cavity, force plug or bottom plate as the case may be. These pins may also be mounted in a separate "pin plate." When so mounted they can be shouldered and riveted without any adverse results.

Where inserts have an interior tap, the mold pin may have a threaded end to receive the insert, or a spindle end which is forced into the tapped hole. This latter method is favored by molders as it is cheaper than using the threaded mold pin. This method, however, has the disadvantage of injuring the threads of the insert, requiring a retapping operation. As inserts are generally brass, and moreover, as tapped holes in these inserts are usually below ¼ inch diameter, this retapping results in a poor threading job and is highly undesirable for Navy work.

Where insert holding pins have a threaded spindle for holding the inserts these pins, of necessity, cannot be permanently secured to the plates, as this would prevent their removal after the molding operation.

HOLE FORMING PINS (PARALLEL TO DIRECTION OF TRAIN OF FORCE PLUG).

These hole forming pins are of the same category as insert holding pins in direction parallel to the direction of train of the force plug.

These pins when of ¼ inch diameter or less should not extend through the molded piece to the top force, but should be terminated flush (minus 0.005 inch) with the surface of the top force when in its final closed position. Larger pins may extend through, if desired.

The diameter of these pins used with C. F. I. should be not less than 3/16 inch diameter. For C. F. G. the minimum diameter of pin may be ⅛ inch.

INSERT HOLDING PINS FOR C. F. I. (90 DEGREES FROM DIRECTION OF FORCE PLUG).

Side insert holding pins must be held securely in the mold by threads or lockrings and they must be so assembled in the mold that they may be readily removed in order that the molded piece may be ejected from the cavity. The minimum diameter of these pins should be ½ inch.

These pins should be positioned so that there will be a clearance between the inner face of the insert and the train of the force plug in order to preclude the possibility of the force plug striking these inserts during the molding operation.

HOLE FORMING PINS (90 DEGREES FROM DIRECTION OF FORCE PLUG).

These hole forming pins are of the same category as the insert holding pins in direction 90 degrees from the direction of travel of the force plug.

These pins should be positioned in the manner described for insert pins so far as concerns clearance between the end of these pins and the travel of the force plug.

KNOCKOUT MECHANISM.

Hand molds, other than those types having stripper plates, are parted and the pieces ejected by means of suitable knockout plates.

Semi-automatic molds and stripper plate molds have their knockout mechanism formed as an integral part of the mold assembly.

Knockout plate should be of steel as specified for the force plate and the knockout pins should be of the same material as the guide pins. Automatic ejection mechanism parts should be steel in all cases, and of a degree of hardness best suited to the particular design and piece concerned.

The thickness of knockout plates should be not less than ½ inch, and the knockout pins should be shouldered and flush riveted to these plates.

All knockout plates, tools and mechanisms should be considered as parts of the mold itself for Government purchase and should be delivered with it.

Hand molds should always be opened and the pieces ejected by means of the knockout plates supplied with the mold. The mold should not be hammered, dropped or bounced to accomplish the ejecting of the pieces.

SPACER FORK.

The use of the spacer fork is previously described under the Flash Filler Plate Mold. This fork should be of machine steel and need not be hardened.

INSERTS.

Provision should be made for the retention of inserts in the mold in such a way that their exact alignment and correct spacing is preserved in the finished part.

It has been found more economical in the molding industry to use inserts of the closed end type, than to use the cheaper type of insert having through tap. With open end inserts, the material will flow around the threads, necessitating a cleaning operation that must be done with a tap. This retapping operation cuts away or enlarges the threads making a loose fit with the machine screw which is undesirable. It is more economical to tap on an automatic screw machine than tap by hand. All Bureau design inserts are now of the closed end type wherever practicable.

Drawings of molds should include in the Bill of Material mention of the necessary metal inserts required for molding the piece.

These drawings should also include in the details of insert holding pins, notes referring to the inserts (threads, etc.) which they hold.

Molds constructed on Government contracts become the property of the Government and are loaned to contractors for production of molded pieces on future Government contracts. It is therefore essential that molds be designed for mounting in any type of press in general use in the molding industry.

As the best and most economical type of mold will depend upon many considerations, such as the characteristics of the raw compound, the size and shape of the molded part, the ultimate use of the molded part, the number of parts to be molded, etc., it is necessary to exercise considerable judgment and careful supervision in the selection of the proper mold for the individual case.

For the check of the completed mold, it is required that at least two samples from each mold cavity be checked for dimensions and fit. Approval of these samples is required to be obtained prior to the approval of the mold and prior to proceeding with production of the order.

In addition to the dimensional check of sample molded pieces, the molder is required to submit, also prior to proceeding with the production of the order, samples of molded pieces in number as required by the particular order, representative of the molded pieces proposed to be furnished under the contract, for tests to determine the compliance of the molded material with the applicable specifications, and to determine the suitability of the mold to produce the proper molded pieces. Samples are required to be taken from the first three heats run from the mold; in the case of single cavity molds, three samples are required; in the case of multiple cavity molds, one sample from each of the four corner and two center cavities of the mold must be furnished for each of the first three heats. Samples are required to be suitably marked for identification, and information submitted in connection with these sample pieces as follows:

Type and manufacturer of insulating material.
Temperature under which samples were molded.
Pressure under which samples were molded.
Time of cure of molded samples.

Approval of these samples is required to be obtained before proceeding with production.

Molds are required to be packed for shipment in individual boxes of standard construction, which will serve for repeated shipments. These boxes are also used for the purpose of stowing the mold when not in use. Individual parts of the mold are required to be coated with a suitable rust preventive compound, and protected from moisture, dirt, etc., by a tough paper or cloth wrapping. These shipping containers are required to be marked with the following data:

Drawing number of mold.
Drawing number of the piece produced by the mold.
Name of the manufacturer.
Contract or order number.

In the case of the Government-owned molds, a very definite need exists for record drawings which will serve for identification, inspection and repair. Before molds are manufactured, assembly and detail drawings of the equipments are required, which will show the following:

Assembly views of the mold.
Details of all principal parts of the mold.
Complete bill of material, giving the name and number of all parts of the mold and showing the materials used therefor.
Finish.
Method of hardening steels.
Weight of mold.

After approval of the construction plans of the mold, finished plans in the form of tracings are required to be supplied.

A complete log is maintained for each mold, giving particulars as to its shipments, use, production quantities, damages, replacement parts, etc. Molds at times have been found to be subjected to more than what may be considered normal wear. Such hard usage of molds results in severe dents, cracks and broken or missing parts. To guard against molds being returned in unsatisfactory condition, they are given a careful inspection after their use on each successive order.

The maintenance of molds in proper condition for issue may be considered of equal importance to the supervision exercised over their basic designs, for if the former is not enforced the latter becomes of little value.

In reviewing the foregoing discussion surrounding the phenolic mold situation, three very pertinent conclusions may be formed which may be of interest to emphasize. First, that the design of these molds, so far as concerns equipments purchased by the Government, is being given careful attention. Second, that attempts to obtain some standardization of these designs have passed through the initial effort, and are now productive of satisfactory results. Third, that through the medium of suitable inspection, the maintenance of molds has been raised to a point where these equipments when furnished to contractors, may be expected to be in satisfactory operating condition.

With the above conclusions in mind, and with a knowledge of the exceptional progress that has been made within the short space of time that this design problem has been active, it can be safely asserted that many of the difficulties experienced with the production of molded phenolics, may be now catalogued as " past history."

NOTES.

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HIGH SPEED TURBINES.

This is the abstract of a paper delivered by Mr. L. Hausman, Northwestern Power Engineer of the Westinghouse Electric and Manufacturing Company, Chicago, on January 11, 1937, and published in the Journal of the Western Society of Engineers in February. Mr. Hausman's diagram of progress in Stationary Plant practice is especially interesting, and he discusses the turbine development made possible by the recent development of 3600 R.P.M. alternating current generators in large ratings, and the problems encountered in building large sets to operate at high pressure and temperature.

During the period 1932 to 1936 very few steam turbine generating units were purchased by the central station industry. The reasons behind this unprecedented long period of inactivity of course are well known. Even after definite signs of improved economic conditions appeared there was a hesitancy on the part of utility executives in applying capital to new undertakings in the form of additions to generating capacity. This period was trying indeed for the manufacturers of steam turbine generators. The designing and building of this class of equipment is, of course, highly specialized and an organization to perform this function requires many years to build up to a state which will make it possible to serve the industry effectively.

The manufacturers wisely carried the burden of keeping together their engineering and manufacturing organizations during the period of inactivity and the Westinghouse Company applied these facilities to the problems which they knew would confront them when the inevitable return to normalcy would occur. They recognized moreover that the very forces which made utility executives, in the face of evident improved conditions, hesitate to make new investments would cause them to judge more critically than ever each new project from the viewpoint of capital investment.

The successful operation of a number of superposed installations for high inlet pressures and temperatures indicated a trend towards this type of unit in plants requiring additional capacity. It is an economical method of applying the technical advancements in steam engineering to existing plants.

Surveys of many plants and inquiries from purchasers clearly showed that the superposed unit was to become one of the most important types of central station turbines in the future and that there would be a demand for these machines in increasingly large capacities.

Having the knowledge of the future trend a development program was established which has led to the design of large steam turbine generating units to operate at 3600 R.P.M. for direct connection to 2-pole, 60-cycle generators.

The desire to extend the design of 3600 R.P.M. turbines into the range of capacities heretofore operating at a speed of 1800 R.P.M. with 4-pole generators is twofold.

The first and more important is the necessity of keeping physical dimensions of the cylinder structure to a minimum in order to assure the maximum reliability in operation. In order to obtain the desired efficiencies with the high steam pressures and temperatures employed, clearance between rotating and the stationary members must of necessity be reduced to very small values. It is obvious that if the masses are reduced to the least value, the possibility of distortion and resultant interference with satisfactory operation will also be reduced to a minimum.

The second and lesser reason for the use of high rotative speeds is the reduction in cost and space requirements in the generating room. The small space requirement in many cases becomes a very desirable attribute since

considerably larger capacities may be installed in a given space by using higher speeds than is possible with low speed units.

The development of high speed turbines as a matter of course had to be paralleled by corresponding development in the high speed turbo generators and the electrical designers deserve a great deal of praise for their part in this undertaking. By 1936 the design of 3600 R.P.M. generators had progressed to the point where units of 50,000 Kw. rating could be offered and in the early part of that year one such unit was sold to an eastern utility to be driven by a corresponding capacity non-condensing superposed turbine designed to operate with 1200 pounds throttle pressure, 900 degrees F. total temperature exhausting at 200 pounds gauge back pressure.

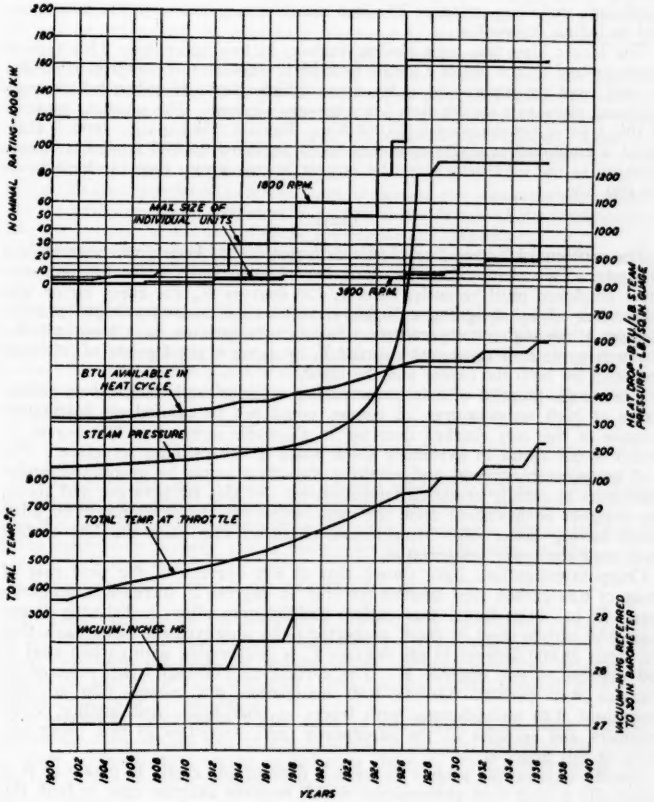
The great strides made in the development of 3600 R.P.M. units are shown graphically in the Diagram. An inspection of the upper portion of the chart indicates that up to 1926 the largest 3600 R.P.M. turbine generator had a nominal rating of 6000 Kw. By 1929 units of 10,000 Kw. rating were built. In 1932 an 18,000 Kw. superposed unit was installed and from this capacity the tremendous increase to 50,000 Kw. was made in one step.

The first 50,000 Kw., 3600 R.P.M. unit built will be placed in operation shortly. Aside from the large capacity it is noteworthy because of a unique innovation in the form of a feed heating turbine, coupled to the main turbine, designed for 200 pounds gauge steam pressure and arranged to exhaust at 5 pounds gauge back pressure with a bleeder opening at 60 pounds gauge pressure.

Many of the condensing turbines over which units have been superposed were built before the adoption of the now customary regenerative feed water heating cycle, and it is found somewhat inconvenient to modify them for extraction service to take care of feed heating in successive stages. The plant in which this turbine is to be installed was thus handicapped. A very satisfactory and efficient heat balance was worked out by the ingenious use of the coupled feed heating turbine. Steam is delivered to the main turbine at 1200 pounds gauge pressure, 900 degrees F. total temperature, and it is exhausted at 200 pounds gauge pressure which is the operating pressure for the existing turbine units. A portion of the exhaust steam is used in the high pressure heater and another portion is delivered to the feed heater turbine where it is expanded to 5 pounds gauge pressure, a portion being taken off at 60 pounds gauge pressure into the intermediate heater. The main turbine passes approximately 1,000,000 pounds of steam per hour, 85 per cent of which is used in the low pressure condensing turbines and 15 per cent is used for the feed heating cycle. The condensate from the low pressure turbines is returned at 150 degrees F. It is then heated through the cycle shown, to 377 degrees F. before being returned to the high pressure boiler. The main unit may be operated without the feed heating element in service if necessary. The entire heat energy supplied to the system by the steam appears either as generator output or in the form of heat in the steam to the condensing units or heat to the boiler feed, with the exception of approximately 3 per cent of the total which is consumed by electrical and mechanical losses.

A more conventional type of superposed unit is rated at 35,000 Kw. designed for 1215 pounds, 900 degrees F.—275 pounds back pressure. This turbine is designed to pass approximately 1,000,000 pounds of steam per hour, but due to the lower heat drop the output is considerably lessened.

Such a turbine will be limited in capacity mainly by the maximum steam flows which can be handled with practical forms of inlet arrangements. A tentative limit of about 2,000,000 pounds of steam per hour has been set which with present operating conditions should be capable of developing capacities closely approaching 75,000 Kw. Generators of this rating for operation at 3600 R.P.M. are now available.



PROGRESS IN OPERATING CONDITIONS.

CONDENSING TURBINES.

Paralleling the development of high speed superposed units a standard line of 3600 R.P.M. condensing units has been developed ranging in capacity from 10,000 Kw. to 50,000 Kw. These turbines are available for any steam pressure up to 1200 pounds per square inch and temperatures up to 950 degrees F. maximum, including swings. The line uses single cylinder construction up to and including 20,000 Kw.

The larger sizes are built tandem fashion in two cylinders. This type is characterized by the use of a single flow high pressure turbine element similar to that used on superposed units, from which steam is exhausted through overhead pipes to a double flow low pressure element. The ultimate capacity of this type of construction is 50,000 Kw. Beyond this capacity 1800 R.P.M. speed is employed in single cylinders up to 100,000 Kw. and tandem arrangement up to the maximum size, the largest in this group thus far built being 165,000 Kw.

"CREEP."

The problem of "creep" or plastic deformation has been given a great deal of study. This phenomenon has long been known and for the top temperatures employed until recently, namely, 850 degrees F., the creep factor was taken care of by designing cylinder structures for correspondingly reduced stresses at the higher temperatures. Satisfactory turbines have been built for steam temperatures up to 850 degrees F. by using a good grade of 0.35 carbon steel for both the casing and the rotor.

The simple practice of merely increasing cylinder wall thickness to reduce stress at high temperatures, of course, could not be carried on indefinitely because of the very marked decrease in allowable stresses at 900 degres F. which would result in extremely thick walls in the cylinder structure. The wall temperature stresses and resulting distortion would be prohibitive under conditions of rapidly changing temperatures. At this temperature and above, the designer is dependent upon the metallurgist for the development of alloy steels having better high temperature characteristics than the low carbon steel used for lower temperature.

Creep investigations have shown that at 850 degrees F. the time rate of creep of 0.35 carbon steel doubles for only 12 degrees F. increase in temperature. It has been found that carbon molybdenum alloy is distinctly lower than 0.35 carbon steel in creep properties; the improvement being such that the latter at 900 degrees to 950 degrees F. is comparable with carbon steel at 800 degrees to 850 degrees F. The carbon molybdenum steel referred to contains 0.43 carbon, 0.40 to 0.60 manganese, 2.5 nickel, 0.30 to 0.60 chromium, 0.20 molybdenum, with traces of phosphorus and sulphur. The cylinders and spindles of the superposed units described in this paper are made of this material.

Specific data on the subject of creep is difficult and costly to obtain as it is essentially a long time phenomenon which requires extreme care in both the control of the test equipment and in the interpretation of the results obtained. Many attempts have been made to interpret short time creep tests for the purpose of extrapolating to long periods, but as yet none of the proposed methods has been generally accepted.

Some idea of the difficulty of obtaining allowable stress data may be learned from the fact that the steam turbine designer is interested in knowing, for example, the stress he may apply to a material in order not to have its plastic strain exceed .001 inch per inch in 100,000 hours operation. Fortunately only the more important parts of the turbine need be designed for this extremely low order of strain.

CYLINDER CONSTRUCTION.

In general the important requirements for satisfactory cylinder designs are uniform heating of the structure as steam is admitted upon starting, absolute freedom from external pipe strains and freedom from distortions which may retard proper movement of the cylinder with changes in temperature due to load changes. The entrapped steam between the cylinder and governing valves must, of course, be kept at a minimum because of the low inertia of the relatively small structure due to the high rotative speed.

In order to provide for uniform heating of the cylinder upon starting, the first nozzles are supplied by steam from the inlet valve through a belt extending around the entire circumference of the cylinder; the inlet valve being located in the cover and the corresponding nozzles in the base. Thus the passage is filled with high pressure, high temperature steam maintaining equal temperatures in base and cover.

Pipe strains are reduced to a minimum by casting the steam chest integrally with the cylinder cover. Great care is used in the design of the lines leading from the throttle to the steam chest portion of the cylinder.

The valves are of unbalanced plug type and are moved by a bar lift mechanism similar to that which has been used for a number of years on smaller units.

The cylinder is supported by arms resting on separate pedestals, the arms, however, merely act as supports. Axial movement of the cylinder is transmitted to the pedestals by a pin joint and draw bar arrangement, so placed as to exert the force in the plane of sliding.

Entrapped steam is reduced to a low value by the integral steam chest construction.

ROTORS.

In the design of rotors, joints are avoided wherever possible. All of the high temperature machines use rotors of solid one-piece construction.

In the case of low pressure rotors for condensing units, discs are shrunk on the rotor body to carry one or more rows of low pressure blading.

BLADING.

The latest practice in the mechanical construction of blading includes the use of shrouded blades throughout the machine. This construction to a large extent eliminates the need for lashing wires. Where the blade heights are such as to make lashings necessary, steel lashing of streamlined section is welded electrically to the blades.

On extremely high speed blades such as are employed on large capacity condensing units stellite shields are attached to the inlet edges of the blades by silver soldering as a means of reducing erosion. This system of protection has proven to be extremely dependable and has the advantage of being easily replaceable.

CONTROL.

With the reduced masses in the rotor structure of high speed turbines the problem of a satisfactory governing system becomes one of considerable magnitude.

A new type of hydraulic governor known as the "Pressure Transformer" type is being applied as standard equipment to all turbines rated at 10,000 Kw. and above. This system consists of essentially three parts, namely, the

reverse flow impeller attached to the turbine shaft, a pressure transformer and a servomotor for operating the inlet valves through the bar lift device.

The impeller is mounted directly on the turbine rotor shaft and delivers oil to the transformer at a pressure which varies directly as the square of the speed of rotation, hence changes in the discharge pressure are an accurate measure of changes in rotative speed.

The changes in discharge pressure with varying speeds provide the governing impulse to the pressure transformer.

The transformer serves to multiply the relatively small changes in impeller discharge pressure so that the total change in fluid pressure available for operating the servomotor relay is correspondingly large and unavoidable friction of relay linkage, etc., is easily overcome, thus giving the system an extremely fast response to changes in load.

A supply of oil is delivered to the servomotor by a second impeller, also mounted on the shaft, thus unlike former impeller type governors, the governing impeller is not used to supply motive oil so that its discharge pressure is not influenced by changes in oil flow as required by the servomotor. The entire mechanism thus provides a control which is simple yet extremely sensitive. Other salient features are as follows:

Elimination of governor drive arrangements such as worms and gears.

Regulation easily adjustable from 2 per cent to 8 per cent.

Speed changer capable of operating over range of one-third speed to full speed.

Exhaust pressure regulation may be obtained without use of external linkage.

CONCLUSION.

Several units of the types of turbines described will be placed in service during this year. At no time during the history of the development of the steam turbine in this country, has the purchaser had the benefit of so much research and development work to serve as a background for a new and important advance in the art, as in these high speed turbines. The future operating records of these machines will reflect the careful and patient work of the design engineers in the many details of construction.

FUEL FOR THE NAVY'S DIESELS.

This is an article prepared by Rear Admiral H. G. Bowen, U. S. Navy, Chief of the Bureau of Engineering, Navy Department, and published in the July, 1937, issue of Motorship and Diesel Boating. It deals with the investigation which has been carried on for some years at the Naval Engineering Station, Annapolis, Maryland, which has now eventuated into the formal specifications under which all Diesel oil for the Navy is purchased, except that for aircraft; and discusses the salient points in the specifications.

The first practical use of Diesel engines in the United States Navy was for the propulsion of the E-class submarines, completed in 1909. Twenty-five years of experience, design and manufacturing development within the Navy, together with the impetus of commercial applications in the marine, railroad, and automotive fields, brought the Diesel engine to the stage where its adoption for purposes other than submarine propulsion was warranted, and justified the present naval policy of installing Diesel engines in all power boats

and in cruisers, destroyers, and aircraft carriers for driving emergency and auxiliary light and power generators. As a result the Navy is today one of the largest single users of Diesel horsepower in the world.

During the first 20-year period of Diesel usage in the Navy, in which the engines were of the comparatively slow-speed, air-blast injection type, the problem of fuel supplies was never a serious one. With the advent of the high-speed, hydraulic-injection engines it became apparent that Diesel fuel specifications must be drastically revised if the performance of the engines was to be maintained at the high standard of reliability and economy in which naval effectiveness has its being.

In January, 1934, the Bureau of Engineering directed the Naval Engineering Experiment Station at Annapolis, Md., to undertake an investigation of Diesel fuels, for the purpose of establishing a standard specification which would guarantee the supply of a satisfactory fuel for engines now in and immediately contemplated for the naval service.

The preparation of satisfactory specifications has required the consideration of practically every definable property of petroleum derivatives, the selection of those properties having major significance with respect to the basic requirements of Diesel fuels, the establishment of acceptable limits in the absolute value of each selected property, and in some cases the development of entirely new test procedures for evaluation of the fuels in terms of the specification.

BASIC REQUIREMENTS.

The basic requirements of a fuel acceptable for naval use were laid down as:

(*a*) A single Diesel fuel suitable for all purposes except aviation engines must be specified. This requirement is dictated by limited tankage accommodations on board ship and simplification of supply and distribution.

(*b*) The specified fuel must be safe and stable in storage. It must be non-corrosive to materials in use for tanks and piping. It must be capable of being pumped at winter temperatures without preheating.

(*c*) The specified fuel must insure engine performance of the highest order with respect to starting, specific fuel consumption, smokeless exhaust, and smooth running under both summer and winter weather conditions.

(*d*) The specified fuel must minimize lay-ups for cleaning and repair; that is, it must not cause deposits on injectors nor in combustion chambers, it must not cause sludging of the lubricating oil, it must not contain corrosive nor abrasive impurities nor produce them as products of combustion, and it must not contribute to mechanical failures by reason of repeated severe combustion shock.

(*e*) The specified fuel must be possible of production by ordinary refining processes from the widest variety of domestic crudes, in order that availability of supplies, particularly in time of national emergency, may be assured.

(*f*) The specified fuel should be possible of production at non-premium prices, unless the military advantages of a premium-priced fuel should prove so outstanding as to outweigh cost considerations.

To date more than one hundred different fuels, selected to represent the major geographical petroleum producing areas of continental United States and a variety of refining methods, such as simple fractional distillation, cracking, and hydrogenation, have been tested at the Engineering Experiment Station. These fuels have been subjected to exhaustive physical and chemical examination and have been run in from two to four types of engines representing both two- and four-stroke cycle operation in both open and air-cell

type combustion chambers for periods ranging from eight to more than one hundred hours each. Different products from the same fields have been blended with each other and cross-blended with similar and dissimilar fuels from other fields. Chemical agents have been added to depress pour points or increase ignition qualities. In each case where fuels have been "doped" or blended, re-tests have been made to evaluate the effects of such treatments.

SPECIFICATIONS.

As a result of these tests, Navy Department Specification 7-02c, dated August, 1936, has been approved for use in purchasing all Diesel fuel except that for aircraft. The detailed requirements of this specification are given in the table.

Test	Test Number		Limit
	Federal Specification Board	American Society Testing Materials	
Flash point, closed cup, °F., min.	110.22	D93-22	150
Pour point, °F., max.	20.14	D97-33	0
Viscosity, SSU, at 100° F.	30.43	D88-33	35-45
Water and sediment, %, max.	300.32	D96-30	0.05
Total sulphur, %, max.	520.22	D129-33	1.00
Carbon residue, %, max.	500.13	D189-30	0.20
Ash, %, max.	542.10	.....	0.01
Corrosion at 212° F., copper strip.	530.31	.....	Negative
90% distillation temperature, °F., max.	100.14	D86-30	675
Diesel Index Number, min.	.....	.....	45
A. P. I. Gravity	40.10	D287-33	..
Aniline point	310.11	.....	..

Each item of the specification was chosen to control with respect to one or more of the basic requirements of a satisfactory fuel, namely:

(*a*) The high flash point is necessary for safety in storage and operation In many states 150 degrees is the legal limit.

(*b*) The low pour point is required so that a single fuel may be stocked for both summer and winter use. Commercially this is not necessary, since production is seldom more than a few weeks ahead of consumption. For military use it is essential, in order that reserve stocks at outlying stations may be suitable for year-round use and on board ships which may change from summer to winter weather conditions within a few days due to change in latitude.

(*c*) Fuels with viscosities below 35 seconds tend to increase fuel pump and injector wear because of their lubricating dryness. With running fits in these parts measured in millionths of an inch, only slight wear is required to increase leakage to the point where delicacy of metering vanishes. Viscosity is also intimately related to the characteristics of fuel spraying, and the upper limit marks that above which a change in the injection system would be required to compensate for the increased penetration and larger drop size accompanying further increases in viscosity.

(*d*) The water and sediment contained in the fuel is a measure of its physical cleanliness. It is axiomatic that freedom from pump and injector trouble and low rates of engine wear depend on the best obtainable physical cleanliness.

(*e*) Sulphur content, ash, carbon residue, the corrosion test, and distillation range are all indicative of the chemical purity of the fuel. High sulphur content produces acid constituents in the exhaust gases which, when reduced below the dew point in an idle engine and continuously on the ship's structure contiguous to the exhaust outlet, result in destructive acid corrosion. Ash formed during combustion is apt to be abrasive in nature and result in rapidly accelerated engine wear. The corrosion test is for the protection of tanks in which the fuel is stored and of the piping and pumps which serve the engines. The carbon residue and distillation range will in general detect fuels which tend toward incomplete combustion and the deposition of unburned carbon, gums, or lacquers on spray nozzles, valves, cylinder heads and pistons. These items also warn against fuels which produce a smoky exhaust, which for naval reasons, particularly in submarines, cannot be tolerated. Furthermore, the distillation range is a measure of volatility, which in turn is one of the controlling items of ignition and burning qualities.

(*f*) Sufficiently high ignition quality is required to insure easy starting in cold weather and to limit the combustion shock in running engines to that which can be continuously tolerated without damage to working parts.

IGNITION QUALITY.

The definition and evaluation of ignition quality has been the bugbear of all fuel investigations. This problem does not exist in spark-ignition engines, and in the early slow-speed, air-injection Diesels substantially constant pressure combustion was an accomplished fact. In the high-speed, hydraulic-injection engine, the importance of the delay or time interval between the exit of the first drop of fuel from the spray nozzle and the beginning of combustion became paramount. The fuel system continues to inject during this delay period and the fuel so accumulated in the combustion chamber burns with almost explosive violence when ignition occurs, thus giving rise to the combustion shock termed Diesel knock.

Two general methods of evaluating ignition quality are in use to varying extents today. One set of ignition quality indices is based on physical and chemical tests. Of these, the Diesel Index Number, obtained by multiplying the A. P. I. gravity of the fuel by the aniline point in degrees Fahrenheit and dividing by 100, has had the widest acceptance. It is used in the Navy specification because for ordinary fuels it shows the best correlation with service use, and is the easiest of determination.

The second set of ignition quality indices is based on engine tests. The engine most commonly used is the American Society for Testing Materials-Cooperative Fuel Research engine, converted to Diesel operation. Ignition quality ratings are made either by determining the critical compression ratio, that compression ratio at which the fuel will just ignite, or by any of several methods which depend on some form of measurement of the ignition delay period. Since day to day atmospheric changes make impracticable the expression of ignition quality in absolute units, all engine tests are reported as comparisons between the fuel under test and standard reference fuels. The tentatively adopted primary reference fuels are cetane, a single, comparatively stable hydrocarbon, which because of its excellent ignition quality has been arbitrarily assigned a value of 100, and alpha-methylnaphthalene, a hydrocarbon so difficult to ignite as to merit a value of zero. Describing a fuel as 45 cetane number means simply that in the test engine it gave results *equivalent* to a reference fuel mixture of 45 per cent cetane and 55 per cent alpha-methylnaphthalene.

Speaking generally, the engine tests are superior to the physical-chemical tests. They approximate conditions under which the fuels are to be used.

For "doped" or specially treated fuels, the physical-chemical indices are quite apt to fail entirely. To date, however, those engaged in fuel research work have been unable to agree on a standard test procedure for engine evaluation of ignition quality, and until such a test is available and recognized in the industry, it will be necessary for the Navy to adhere to the Diesel Index Number in its specifications. In the normal range of ignition quality; that is, between about 40 and 55, the cetane and Diesel Index Numbers are approximately equal.

GENERAL RESULTS.

Specifically, it is not expected that any of the 15 or 20 test procedures now in use in different laboratories throughout the country will be developed to the point of exact correlation with large engines in field or service use. As noted, these methods depend entirely on some form of measurement of the ignition delay period. While it is true that ignition delay is the underlying cause of Diesel knock, net effects occurring after ignition vary with different fuels. Although the first phase of combustion is described as approaching explosive violence, actually different fuels show such difference in burning rates after ignition that the knocks occurring in service engines may appreciably differ from those predicted from measurement of the delay period only in the laboratory.

The Engineering Experiment Station has accumulated sufficient evidence to demonstrate that fuel quality ratings should be based on the net shock effect on the running engine, and has developed a method of evaluating fuels by their shock effects. The method is applicable to any engine either in the laboratory or in the field and requires the use of only those instruments necessary to obtain pressure-time diagrams, which are available to most engineers. However, the measurements and calculations involved are quite tedious, and the method, while correct in principle and practicable in service, is too cumbersome and expensive for ordinary use in routine testing of contract fuels.

Because of its potential value to the nation as a whole, as well as to the Navy, the Diesel fuel investigation at the Engineering Experiment Station has not been held confidential. Engine builders, fuel refiners, and the scientific societies engaged in similar research have cooperated in this investigation.

The problem has not been solved to the complete satisfaction of all concerned, and even if it were, the cry for higher powers on less weight in smaller spaces would be heard again. Future development of the Diesel engine and its fuel must go forward together.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

THE ADMIRALTY TYPE TEST FOR COMPRESSION-IGNITION ENGINES.—The Oil Engine, London, England, July, 1937.

Every oil engine purchased for Admiralty service has to be of a type of which the performance characteristics have been accurately determined and can bear comparison with those of other engines of its class. A test programme recently prepared provides for the complete testing of one engine of a particular design in normal production and the results obtained are taken as a standard for succeeding engines of similar size and of that design. The

tests are of a severe nature, amounting to a complete re-rating of the engine by the Admiralty experts, and manufacturers are prohibited from disclosing the fact that an engine has passed or has been subjected to them. This type test is carried out either at the maker's works or at an Admiralty establishment, and in the former case the test is witnessed by an Admiralty representative.

Whenever practicable, the engine to be tested is selected by an inspecting officer from a batch of complete engines that have been made in the normal course of production, and after this selection of an engine no work is done on it, except as is in accordance with the requirements of the type test. Before starting the tests, the engine builder must demonstrate that the test plant is in a reliable and satisfactory condition, and that it is capable of fulfilling the following general requirements: Variation of load and R.P.M. without stopping the engine; measurement of the B.H.P. developed by means of the torque reaction of a suitable dynamometer; accurate measurement of the fuel consumption; measurement of the R.P.M. by counter or similar device; adequate provision for the determination and control of temperatures and pressures; visual inspection of the exhaust gas shade.

Before the engine test begins, accurate measurements are taken from cylinder liners, pistons and rings, crankshaft and bearings for comparison with readings taken after the tests.

Loop tests, that is, tests to find the variation in fuel consumption with variations in load, when the engine is running at constant speed, are carried out to cover a range from 20 per cent of the normal B.M.E.P. recommended by the maker for continuous running at the speed in question, up to the highest load at which the maker considers it safe to run the engine for the necessary test period. At least six sets of measurements are required to define each curve, and five loop curves are normally required at specified speeds varying between 40 per cent and 100 per cent of the maximum speed of the engine. In the case of an engine where pressure charging is employed, additional loop curves may be required at various boost pressures. Each test has to be of sufficient duration to ensure an accuracy of plus or minus 1 per cent in the measurement of the weight of fuel consumed. At each speed, loads are progressively increased from the minimum to the maximum, the results obtained being checked by repeating the series in reverse order. On completion of these tests, the loop curves and data of exhaust shades and temperatures are submitted to the Admiralty, and from this information and appropriate power and speed (Admiralty Test Rating) is assigned.

Carrying 95 per cent of the A.T.R. torque, the engine is run at the A.T.R. speed for an endurance test of 72 hours either continuously or in periods decided upon by the Admiralty, according to the service for which the engine is contemplated. Any stoppage which occurs during this test must be fully reported. It is followed by a full-power test during which the engine is run continuously for a period of 12 hours at the A.T.R. power and speed. After this comes an overload test in which an engine intended for service at constant speed is run for two hours at the A.T.R. speed, carrying 110 per cent of the A.T.R. torque.

An engine intended to run in service at constant speed, *e.g.*, driving an electric generator, is subjected to a governing test with the driven unit coupled up. On this test, the temporary and permanent changes of speed with changes of load are measured when the full rated load is suddenly thrown on or taken off, and also when the load is changed gradually or by steps not exceeding 20 per cent of the rated load. Starting and slow-running abilities are noted and tests to obtain torsional vibration records may be conducted.

Should any vital part of an engine fail completely during the course of

the tests, this happening is held automatically to reject the engine. After renewal of the part or parts, the engine may, at the discretion of the Admiralty, be resubmitted to tests exactly as a new unit.

On the completion of the tests, the engine is dismantled, and during this work features regarding its accessibility are noted by the Admiralty inspecting officer. The parts are laid out for inspection in the condition in which they were removed from the engine, and measurements are taken from the same positions at which the initial gaugings were made. A detailed report of the condition of the principal parts, the hours of running since the previous examination, and details of any wear revealed by the gaugings, are included in the data on which the type test report is based. A copy of the report is supplied to the engine builder for his information, and on the "type" engine's performance as disclosed by the report, the Admiralty decide whether or not such engines will be purchased.

CONTROL OF PITCH DIAMETER OF BOLTS AND NUTS TO BE GALVANIZED.—The Bureau of Standards Technical Bulletin, July, 1937.

From a study of the change in pitch diameter of bolts and nuts on galvanizing, and an analysis of the National Screw Thread Commission pitch diameter tolerances for classes 1, 2, and 3, Henry W. Bearce, co-chief of the Bureau's Weights and Measures Division, concludes that if nuts are tapped to pitch diameter limits between class 2 min. and class 1 min., before galvanizing, the bolts and nuts will assemble after galvanizing, without being either too light or too loose.

The use of class 3 not-go plugs and class 2 not-go rings as go-gauges, in preliminary inspection, in order to avoid the purchase of special size go-gauges to be used on work to be galvanized, is suggested.

GERMAN SHIPBUILDING.—The British Motor Ship, London, England, May, 1937.

The extent of the progress made recently in German shipbuilding is not fully realized. On April 1 the total tonnage of ships on order or under construction was 1,098,315 gross, although those under construction, according to Lloyd's figures, aggregated only 352,938 tons gross. The large motor ships (of over 3000 tons gross) amount to 729,200 tons gross, compared with Lloyd's figure of 255,331 tons actually under construction. This large tonnage is on order at the various yards as specified in the particulars given below.

Shipyard.	Number of Motor Ships on Order.	Gross Tonnage.
Deutsche Werft	34	277,200
Bremer Vulkan	15	121,200
Blohm and Voss	4	69,500
Fr. Krupp	7	69,500
Deschimag	10	64,400
Howaldtswerke	4	46,000
Schichau Yard	4	38,000
Flensburg Yard	8	43,400
Total	86	729,200

The actual tonnage of motor vessels is, of course, considerably larger than this, as there are numerous smaller ships being built. Of the total tonnage now on order 56 per cent is for foreign ownership, and the vessels now under contract both in tonnage and number are approximately double those of a year ago.

Germany has more motor ships on order than any other country, including the United Kingdom, and it is worth noting that of the 34 oil-engined vessels under contract with the Deutsche Werft, 27 are for foreign ownership. Most of them are for Norway.

HYDROSTATIC TESTING.—Combustion, New York, N. Y., July, 1937.

A recent newspaper account of a man being killed by happening to be directly in the path of a jet of high-pressure gas suddenly released through a defect in the container, raised the question among a group of engineers as to the possible effect of a jet of water issuing from a high-pressure boiler while under hydrostatic test.

Few new boilers are absolutely tight when first tested hydrostatically, but the leaks around tube ends, handhole plates or gaskets usually develop before a considerable pressure has been built up and are in the form of seepage or minute jets that are soon dissipated. It has been pointed out that, even where a leak develops at high pressure, the capacity of the hydrostatic pump is so small that the pressure is immediately relieved before any appreciable volume of water can be discharged. Of course, the initial impingement of a jet of large diameter might be extremely unpleasant or inflict injury if a person were close and directly in its path, but where such larger jets might most likely occur are in locations where persons are not likely to be.

Viewing the problem from another angle, however, the compressibility of water, as given in the handbooks, is one part in three hundred thousand per pound of pressure exerted, so that under two thousand pounds per square inch the water would be compressed by one-one hundred and fiftieth of its original volume. This compression, disregarding the pump, would act to sustain a jet until the original conditions were reestablished.

Another point that may be raised is the fact that no containers, regardless of the thickness of the walls, are completely inelastic. The elasticity of steel is about one-twentieth of one per cent per thousand pounds pressure and the amount of expansion of a high-pressure boiler drum under two thousand pounds pressure when cold would in its effect be at least equivalent to that produced by the compressibility of the water.

Furthermore, if a considerable amount of air were entrapped within the unit its expansion would serve as a sustaining force for the jet, but care is always taken to vent the air fully as the boiler is filled and that entering with the water is in solution, hence it would exert little influence. Therefore, under the proper procedure, air may be dismissed as a factor.

Obviously, if any metal part were to let go the result might be disastrous, but the care exercised in building present high-pressure boilers and the thorough inspection to which they are subjected renders this possibility very remote.

While an analysis of the problem serves to discount such dangers as may seem apparent at first glance and, despite the absence of accidents from this cause, it nevertheless behooves those not directly concerned with such a test to exercise caution when around a high-pressure vessel that is under test, and especially one should avoid standing in front of flanges and elbows.

TWIN-SCREW TUG WITH KORT NOZZLES.—Shipbuilding and Shipping Record, London, England, July 1, 1937.

The first twin-screw tug fitted with Kort nozzles to be built in Britain was recently completed by Richard Dunston Limited, Thorne, near Doncaster, to the order of the London & North Eastern Railway Company for service in Lowestoft Harbor.

Built to the requirements of Lloyd's Register for towing services, the vessel is sub-divided by four watertight bulkheads into five compartments, comprising the fore peak, crew accommodation and feed tank, machinery spaces, store and after peak.

The principal particulars are:

Length, b.p., feet	71
Breadth, molded, feet	18
Depth, molded, feet	9
Draught, mean, feet and inches	7.4

The stern, designed to suit the Kort nozzles, departs from a normal design in that no stern frame is required and the keel runs up from the after peak bulkhead in a straight line to just below the load water line at the extreme after end. The Kort nozzles are fitted with watertight diaphragms which isolate them from the main hull. Twin rudders are fitted, foot-step bearings being embodied at the bottom of the exit of the nozzles, the rudders being connected on deck by link gear and operate together, controlled by Donkin's hand and steam steering gear from the bridge. Twin rudders have the advantage that steering way is obtained as soon as the engines are started ahead irrespective of whether the vessel has any way on her or not, since rudders lie entirely in the accelerated propeller slip area.

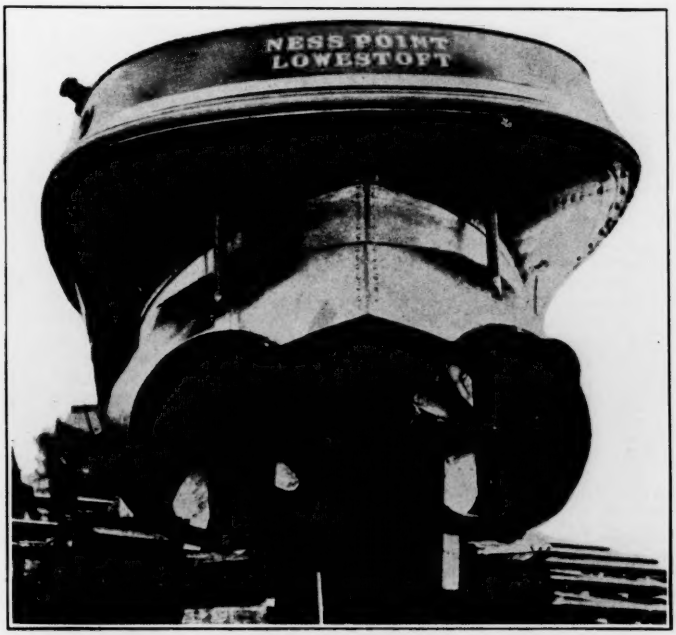
In the original specification, for a tug of normal design, the owners called for 300 I.H.P., a 3 tons pull, and a free running speed of 9 knots. The Kort Propulsion Co., Ltd., London, were asked to submit their assessment of power for a 3 tons pull, and in view of the reduction in horsepower which could be anticipated for the bollard pull by fitting Kort nozzles, they put forward a collective I.H.P. of 180, but assessed a free running speed of only 8 knots.

In view of the reduced horsepower the Kort Propulsion Company suggested to the builders that they could take advantage of a smaller hull, but the owners decided to build the tug of the same dimensions as for the 300 I.H.P. originally suggested, with a reduction in the machinery space and bunker capacity. The nozzles were supplied by R. H. Green & Silley Weir Limited, in their capacity of licensees of the Kort Propulsion Co., Ltd., and were dispatched to the builder's yard complete and ready for fitting to the hull.

Propelling machinery consists of two sets of vertical compound surface-condensing engines built by Plenty & Son, Newbury, which at full power turn 160 R.P.M.

Trials of the *Ness Point* were carried out early in June on the Humber and in the Hull Docks. A standing pull of 4.1 tons was obtained at the designed revolutions of 160 per minute, the horsepower being in the region of 180. The free running speed was 8.3 knots.

With regard to maneuvering ability, with both engines running full ahead and the rudder to starboard, the vessel turned a complete circle in 1 minute 56 seconds, and with the rudder to port in 1 minute 53 seconds, the steering circle being about 150 feet in diameter. With the starboard engine full ahead and the port full astern, helm hard to port, the vessel turned a complete circle in 1 minute 50 seconds, the diameter of the circle being little more than the



STERN OF THE "NESS POINT" SHOWING KORT NOZZLES.

length of the tug and was less than 100 feet. With the tug running full speed ahead and the engine brought to full astern, way was taken off the vessel in 37 seconds.

After trials the tug was maneuvered between two large vessels in dock, the space between them being less than 5 feet in excess of the overall length of the tug. The *Ness Point* was laid alongside the quay in this space without a fender which showed her fine maneuvering qualities.

These trials are interesting and will undoubtedly be of great importance to tug owners and builders and it is certain that the future performance of this vessel will be closely watched.

The specification for the *Ness Point* was prepared by the Engineer, Southern Area, L.N.E.R., on whose behalf the Docks Engineer supervised the building of the vessel.

AGAIN COILED.—The General Electric Review, Schenectady, N. Y., July, 1937.

Perfection of a new high-efficiency tungsten filament, which will increase the light output of incandescent lamps 10 per cent without using any additional electric current, has been announced by J. E. Kewley, vice president of the G-E incandescent lamp department at Nela Park, Cleveland.

The new high-efficiency filament climaxes 24 years of continuous research in the lamp development laboratories, Mr. Kewley declared. First drawn into a straight wire 19/10,000ths of an inch in diameter, so fine as to be almost invisible to the naked eye, the tungsten is then wound, 335 turns to the inch, around a thin molybdenum wire, or mandrel, leaving the coils, or turns, 1/1000ths of an inch apart. These coils must be kept as close together as possible, to reduce heat loss, but must not touch each other, else they would short-circuit, causing the lamp to fail instantly.

Then the coiled wire is coiled once more on another mandrel, 70 turns per inch, with a spacing of 7/1000ths of an inch between the secondary coils. Before the first coiling, the tungsten wire in a filament is 20 inches long. The first coiling compresses it to a length of 3.4 inches. The second coiling compresses it still further to a length of ⅝ inch, with a coil-diameter of 310/10,000ths of an inch. Following the second coiling, the mandrels or center wires are dissolved by means of chemicals.

When installed in the lamp, the new filament is mounted as a cross-bar between two lead-in wires, with one support in the center of the filament, instead of being looped around three supports, as was the practice with the older type of coiled filament.

Dr. Irving Langmuir, of General Electric's research laboratories in Schenectady, in his invention of the gas-filled lamp, discovered that the greater the concentration of the filament, particularly in length, the lower the heat loss. Since the effective length of the new design filament is only about one-half that of the old, there is much less cooling by the gas, which permits the lamp to give 10 per cent more light for the same amount of electric current. The gas employed in the new lamps is a mixture of nitrogen and argon.

CAST CRANKSHAFTS.—Engineering, London, England, May 21, 1937.

The improvement which has taken place in foundry technique in recent years, and more particularly the development, on scientific lines, of high-strength alloy cast irons, has once again drawn the attention of engineers to

the cast crankshaft for multi-cylinder motor-car and other engines. From the points of view of ease of manufacture and lower cost, the cast crankshaft possesses definite advantages over the forged product while, if the material employed be an alloy cast iron, the marked resistance to wear and freedom from notch sensitivity under fatigue stressing, associated with many of these materials, constitute further advantages. The valuable properties of some of the latterly-introduced alloy cast irons, in the as-cast state as well as when heat treated, give an indication of the possibilities likely to result from continued research in this field, the exploration of which has but recently been begun. The recognition of this fact was one of the reasons why an investigation on materials for cast crankshafts was put in hand some time ago at the National Physical Laboratory, on behalf of a co-operative research committee, comprising representatives of the Research Department of the Institution of Automobile Engineers, the British Cast Iron Research Association, and various industrial firms interested in the production and development of cast crankshafts. Other reasons which prompted the investigation were the absence of combined fatigue-stress data relating to the modern cast materials and the development, at the National Physical Laboratory, of an ingenious high-speed combined fatigue-stress testing machine, described in a contribution presented before the Institution of Mechanical Engineers by Dr. H. J. Gough, F.R.S., and Mr. H. V. Pollard on November 1, 1935, and dealt with in our columns at the time.

The investigation, the scope of which was intended to occupy a period not exceeding two years, has actually been completed well within that time, and the results obtained were embodied in a paper, entitled "Properties of Some Materials for Cast Crankshafts, with Special Reference to Combined Stresses," submitted at a recent meeting of the Institution of Automobile Engineers by Dr. Gough and Mr. Pollard. The object of the investigation, it should be emphasized, was to secure accurate data relating to typical cast materials which were either already employed or were in the completing stages of development for the production of engine crankshafts. Five commercial alloys were accordingly selected, comprising a heat-treated cast steel, the percentage composition of which was: carbon, 0.32; nickel, 2.42; chromium, 0.49; and molybdenum, 0.38; a heat-treated 1.75-per cent copper-0.46 per cent chromium iron; an inoculated iron; a 0.42-per cent chromium-0.95 per cent molybdenum iron; and a 1.87-per cent nickel 0.47 per cent chromium iron. The last three materials were supplied and tested in the as-cast condition. Complete static tensile and torsional tests were carried out, and these bring out the essential differences between the cast and heat-treated alloy steel and the four cast irons. Whereas the steel is essentially a ductile material, each of the four irons has no clearly-defined yield points, little deformation at fracture and negligible absorption of energy under the Izod notched-bar test.

The most important section of the mechanical tests, however, relates to the investigation of the resistance of the materials to fatigue stresses, as determined with the aid of the new combined fatigue-stress testing machine. Additional fatigue tests were made in a Wöhler machine and in a Stromeyer alternating-torsion machine, with the result that a somewhat bewildering array of fatigue-limit values, each having its own particular significance, has been deduced. Happily, however, the authors have drawn up a table which shows very clearly the relative fatigue resistances of the five alloys, when tested in the combined-stress machine, for the two cases of reversed bending stresses and reversed torsional stresses. The ultimate tensile strengths of the materials vary widely, by as much as 64 per cent, that of the alloy steel being 52.3 tons per square inch, that of the copper-chromium iron 32.3 tons, and those of the inoculated, chromium-molybdenum and nickel-

chromium irons, respectively, 23.3 tons, 20.8 tons, and 18.8 tons per square inch. Under reversed-bending stresses, however, the relative values of the endurance ratio (obtained by dividing the fatigue limit by the ultimate tensile strength) show no very marked variations, the maximum difference being only 11 per cent. The value of this endurance ratio in the case of the alloy steel is 0.50, whereas in that of both the copper-chromium iron and the nickel-chromium iron it is 0.56, in that of the inoculated iron it is 0.55, and in that of the chomium-molybdenum iron it is 0.51. These values of the endurance ratios, as the authors point out, would all be considered normal for high-quality ductile steels. The torsional endurance ratios in the case of the four cast irons, the values of which range from 0.44 to 0.52, are distinctly higher than the ratio of the alloy steel, the value of which is 0.34. These figures, it is emphasized, indicate that the general behavior of the alloy steel under combined stresses is much akin to that of wrought steel, while the irons develop characteristics more definitely associated with the theoretical behavior of a class of materials of a more brittle nature.

The general conclusion arrived at as the result of the investigation, therefore, is that the fatigue resistance of each of the five materials examined is high in relation to its tensile strength. This, needless to say, is a factor of considerable importance in alloys for cast crankshafts. A further point of interest is that owing to variations from sample to sample and to certain casting defects, both of which factors were accentuated by the comparatively small section of the bars prepared for the tests, it is quite possible that the values obtained do not represent the optimum properties of which the materials are capable, when produced under more favorable conditions and in the form of full-sized castings.

PROTECTIVE SCREENING OF RADIUM DURING TRANSPORTATION.—Bureau of Standards Technical Bulletin, Washington, D. C., August, 1937.

The problem of screening medical radium preparations during transportation has received renewed attention since the recent order of the United States Post Office Department excluding all radioactive preparations from the mails. This exclusion was not ordered to protect personnel, but to protect photographic film, large quantities of which are shipped by mail.

This action caused considerable protest on the part of users of radium and radon, and the Post Office Department requested the Bureau to investigate conditions under which radium might be transported with safety to photographic films. In the course of this investigation which was conducted by L. F. Curtiss, it became obvious that the protection of film imposes much greater restrictions than the protection of personnel, so that the satisfactory solution of the film problem would also result in regulations which would eliminate any chance of personal injury from radium while in transit.

Preliminary exposures of ultra-speed X-ray film to preparations of various radium content through different lead filters at distances ranging from a few inches to several feet clearly revealed that for radium preparations containing 25 to 100 mg of radium no reasonable amount of lead screening alone would protect photographic film, for the length of time they might be together in a mail pouch. As in many other instances, it became evident that the inverse square law must be utilized by restricting the minimum distance between the radium package and the photographic film. Further studies revealed that this minimum distance must be from about 4 to 20 feet,

depending on length of exposure, thickness of lead filter, and strength of radium preparation. Obviously, it would be impossible to use a package so arranged that it would be safe to store photographic film in contact with it.

From these orienting tests it seemed best to first design a package giving reasonable protection to persons handling it, making the package as compact as possible, consistent with such protection. Secondly, figures were secured to show the thickness of lead filter required for different distances, exposure times, and radium content up to 100 mg. These results were plotted, and from the curves thus obtained a table has been prepared for the use of shippers and transportation companies to indicate the amount of lead screening and the minimum safe distance from photographic film, for a given amount of radium for a particular time in transit.

The package used in obtaining these data was designed to insure a thoroughly safe distance between the hand and the radium container. It is about 14 inches in length and 6 inches square, and is arranged to stand conveniently with its long dimension upright. A rigid handle is attached to the upper end. The compartment for the lead cell containing the radium is located centrally near the bottom so that the hand is about 10 inches from the radium. Even if the hand should be placed against the side of the box opposite the radium container it would be nearly 3 inches from it. There is no reason why this should be necessary, and the aim in designing the package has been to make it inconvenient to carry it in any other way than by the handle. Thus, the use of this package in combination with the table of safe distances should make possible the transportation of radium without danger to personnel and goods in transit.

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